SLIDE VALVE GEARS.

*AN EXPLANATION OF THE ACTION AND
CONSTRUCTION OF PLAIN AND
CUT-OFF SLIDE VALVES.*

BY

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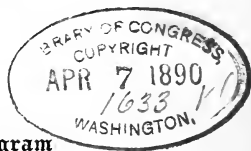*ENGINEER OF THE RAND DRILL COMPANY;*

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𝔄nalysis by the 𝔅ilgram 𝔇iagram

TO

Professor John E. Sweet,

TO HAVE BEEN WHOSE PUPIL I CONSIDER ONE OF THE
GREATEST PRIVILEGES OF MY LIFE,
THIS LITTLE VOLUME IS GRATEFULLY INSCRIBED.

PREFACE.

THIS work has been prepared to meet what the author considers a real want. It has been written with the aim of making it intelligible to any one who might be willing to make a serious effort to understand it. High authority exists for a mathematical treatment of the subject, but with this the author has no sympathy. Designing a valve gear is essentially a drawing board process, and a mathematical treatment of it is simply an uncalled for use of heavy artillery. The graphical treatment is therefore adopted throughout.

Acknowledgment is due to Mr. Hugo Bilgram for his courtesy in kindly permitting the use of his valve diagram. The author has all due respect for the Zeuner diagram, but that respect is not incompatible with the conviction that Mr. Bilgram's method is a marked improvement upon it. Valve diagrams are used for two purposes—to analyze existing valve motions and to design new ones. The Zeuner diagram fulfils the first purpose perfectly, but is unsatisfactory when applied to the second. The leading data that are given in designing a valve motion are the point of cut-off, the port opening, and the lead of the valve (not the lead angle of the crank, as is often conveniently assumed). It is the radi-

cal defect of the Zeuner diagram that none of these dimensions can be laid off from known points. The lead must be laid off from an unknown point of the centre line, and the port opening from an unknown point on an unknown line. Finally, through these unknown points and the centre of the shaft the valve circle is to be drawn from an unknown centre and with an unknown radius. Under these circumstances the result sought is found only through blind trial. With Mr. Bilgram's method all this is changed. The lead is laid off from a fixed line, the port opening from a fixed point, and the cut-off position of the crank is located. The lap circle is then drawn tangent to these lines, and the problem is solved. Moreover, the awkward conception of the backward rotation of the crank is obviated. Finally, these marked advantages are not accompanied by any compensating disadvantages whatever.

Acknowledgment is also due to the *American Machinist* for the use of a number of engravings originally prepared to illustrate some of the author's articles in that paper.

The irregularities due to the connecting rod introduce peculiar difficulties into the study of the first principles of the slide valve, which difficulties were first overcome by the happy expedient of using the slotted cross-head instead of the connecting rod in the preliminary study. For this, together with many other original and highly valuable contributions to the subject, we are indebted to Mr. W. S. Auchincloss, who first published them in his well-known and standard work entitled *Link and Valve Motions*, to which those who wish to prosecute their studies beyond the scope of this work are referred.

The author has gone more fully than is customary into the methods of equalizing the various events of the stroke. The sections relating to these methods will be found more difficult to follow than the others, while at the same time they form no necessary part of a *general* treatment of the subject. Those who begin their studies of valve motions with this book, may find these chapters too difficult for the first reading. They have, therefore, been marked with a star (*) in the Table of Contents and in the body of the book, in order that they may be omitted, if desired, in the first reading; and it should be understood that the chapters not so marked form of themselves a complete connected treatise, of a more elementary character than the book as a whole.

PHILADELPHIA, Oct. 19, 1889.

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Part I.

THE SLIDE VALVE WITH FIXED ECCENTRIC.

The Slide Valve with Fixed Eccentric.

THE PLAIN SLIDE VALVE.

Fig. 1 is a sectional view of a plain slide valve and its seat, the valve being shown in its central position, with the ports completely covered by it.   The distance
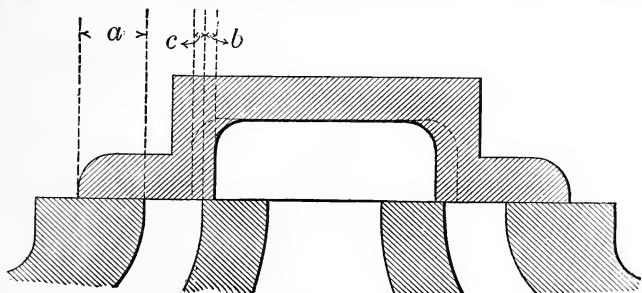
Fig. 1

a, by which the valve extends beyond the steam edge of the port, is called the outside lap, steam lap, or more usually, simply lap of the valve.   The distance b, by which it extends beyond the exhaust edge of the port, is called the inside lap or exhaust lap.*   The exhaust

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* As will be more fully shown later on, valves are sometimes so made that the steam is admitted by the inside and exhausted by the outside edges.   Hence the terms inside and outside lap are somewhat ambiguous.   The terms steam lap and exhaust lap avoid this ambiguity, and are to be preferred.

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lap is always much smaller than the steam lap. It is
frequently absent, and frequently the exhaust edge of
the valve does not reach the exhaust edge of the port,
being made as shown by the dotted lines. In that
case the distance c is usually called inside clearance,
though a better name is negative inside or exhaust lap.
It is sometimes called inside lead or exhaust lead; but
these terms should not be applied here, as they have
properly another definite meaning, which will be ex-
plained farther on. The measurement for both steam
and exhaust lap is made for one end of the valve only.
Thus if a valve is said to have $\frac{3}{4}$ inch lap, the meaning
is that it has that much at each end.

THE ECCENTRIC.

The slide valve is usually driven by means of an eccen-
tric on the crank shaft, and it becomes necessary at the
outset to obtain a clear conception of the motion which
the eccentric gives. In brief, the eccentric is a short
crank with a large crank pin. It is obvious that the
motion of a cross-head would not be changed by in-
creasing the size of the crank pin. If a crank pin were
enlarged until the crank shaft came within the cir-
cumference of the pin, the result would be an eccen-
tric. The arm of a crank is the distance from the
centre of the shaft to the centre of the crank pin, and
similarly the "throw" of an eccentric is the distance
from the centre of the shaft to the centre of the eccen-
tric disc. Usually the centre of the disc is within
the circumference of the shaft; but this does not alter

the nature of the device, which remains simply a short crank with a large crank pin.

THE SCOTCH YOKE OR SLOTTED CROSS-HEAD.

The usual method of connecting the cross-head to the crank pin by means of a connecting rod introduces certain distortions and irregularities into the relative motions of the piston and crank. These will be more fully explained farther on, but it is desirable in the first instance to avoid the necessity for considering them, as they greatly complicate the subject. This is accomplished by considering in the first instance an engine having the piston and crank connected by means of the device called the Scotch yoke or slotted cross-head, since that connection is without the distortions mentioned.* As has been explained, the eccentric is essentially a crank; and it follows that the distortions which are introduced by the connecting rod into the motions of the piston and crank, are also introduced by the eccentric rod into the motions of the valve and eccentric. The reasons which lead to the adoption of the slotted cross-head in place of the connecting rod also require its use in place of the eccentric rod. An engine fitted with slotted cross-heads is illustrated in Figs. 2–11. The slotted cross-head will be recognized at once, and is too familiar a device to need further description.

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* The slotted cross-head is employed here with the permission of Mr. W. S. Auchincloss, to whom the thanks of the author are due.

Fig. 2
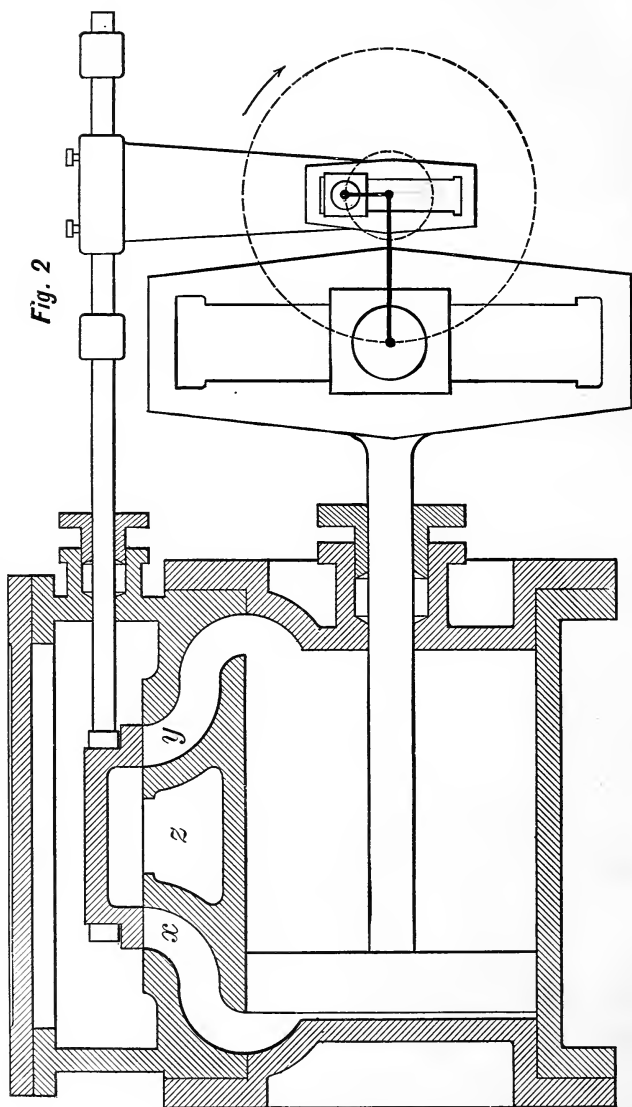
THE PRIMITIVE ENGINE.

When illustrating the action of the valve of a steam engine, it is essential for clearness that the valve be shown on the top of the cylinder. A valve so located in an actual engine would require the use of a rock shaft to communicate the motion of the eccentric rod to the valve rod—a construction which finds use in American locomotives. This rock shaft complicates the action of the parts, and it is desirable in this preliminary work to avoid it. To accomplish this the unmechanical arrangement of Figs. 2–11 is adopted.

Figs. 2–6 represent the primitive engine with the parts in a number of successive positions. The valve has no lap on either steam or exhaust side, and the eccentric is set at right angles to the crank, and in advance of it in the direction of the rotation. The eccentric, being in fact a crank, is represented as such, and the valve is driven from it by a slotted cross-head, which is secured to the valve stem by the bracket shown. In Figs. 3–6 the slotted cross-heads are represented by their centre lines only, for greater clearness and simplicity.

Referring to Fig. 2, the crank is on the "centre," and the parts are ready to begin movement, the direction of rotation being as shown by the arrow. In Fig. 3 the crank shaft has turned through an angle of forty five degrees, carrying the parts to the positions shown. Considering Figs. 2 and 3, it is clear that the first movement of the crank shaft carried the valve to the right, and thereby opened port x to steam and y to exhaust. Opening port x admitted steam behind the piston to

Fig. 3

Fig. 4

Fig. 5
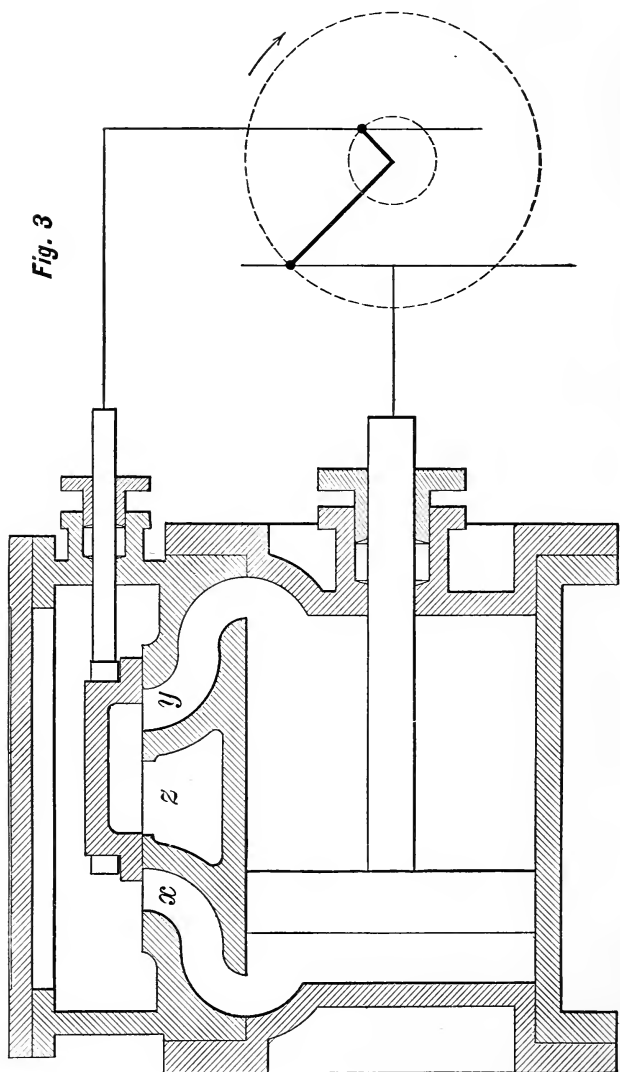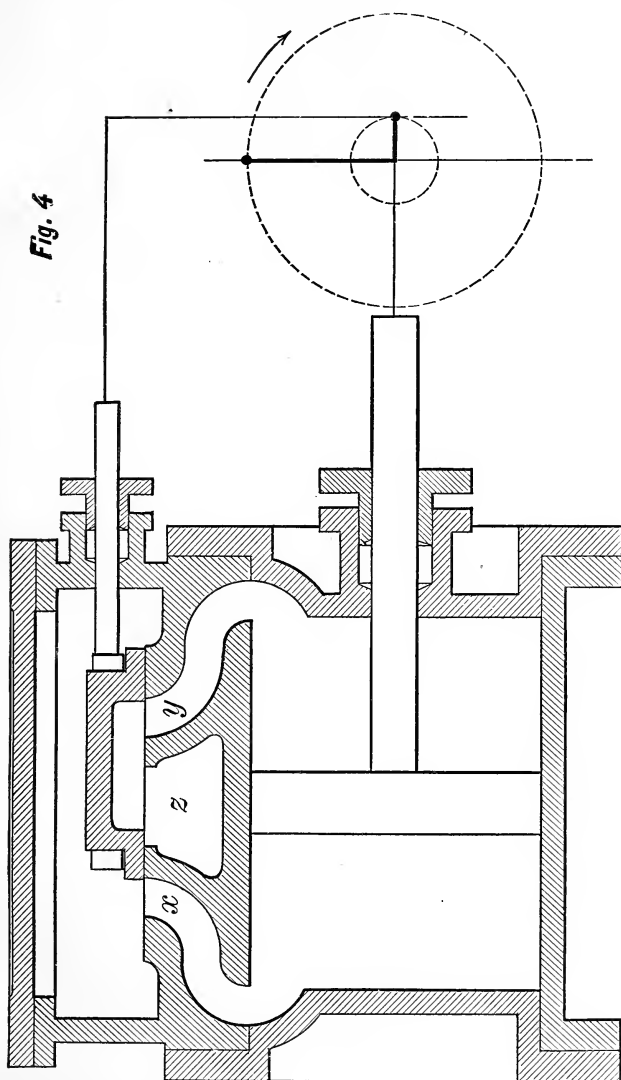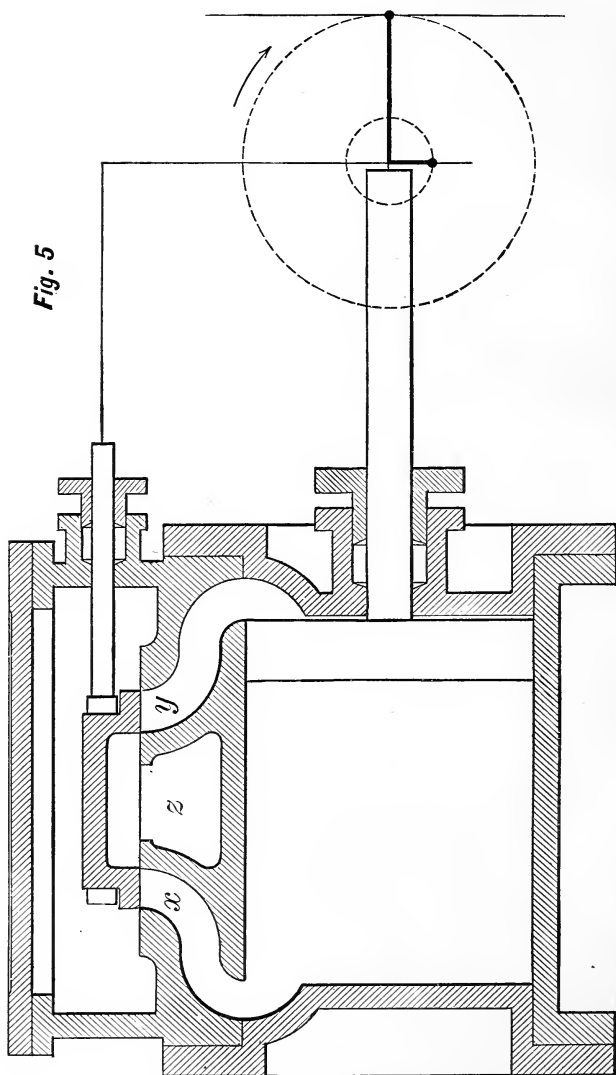
drive it forward, and opening y enabled the steam which previously filled the space in front of the piston to escape to the cavity z, which communicates through the exhaust pipe with the atmosphere or condenser, as the case may be. In Fig. 4, the crank shaft has turned through an additional angle of forty five degrees, bringing it at right angles to its initial position. The piston is now at the centre of its travel, and the valve at its extreme right hand position. As the rotation continues, the piston continues to advance; but the valve reverses its motion, and gradually closes its ports, until when the crank completes a half revolution, as shown in Fig. 5, the valve reaches its middle position at which it stood in Fig. 2, with all ports closed. Continuing the motion, the valve is carried to the left, opening port y to steam and x to exhaust, as shown in Fig. 6, and the piston is driven back to its original position; and this sequence of operations will obviously continue indefinitely.

With the eccentric located as in the figures, the direction of rotation must be as described. This will be apparent if, starting with Fig. 2, rotation in the opposite direction be imagined. The effect of this would be to open port x to exhaust and y to steam, thereby effectually stopping the rotation in the direction imagined. To effect this reverse rotation the eccentric must be located diametrically opposite to the position shown in the figures.* The student should satisfy himself of

* This is true with the primitive form of valve only. With valves having lap, as actually used, the eccentric position for reverse rotation is *not* diametrically opposite from the position required for forward rotation. This subject will be referred to again.

Fig. 6
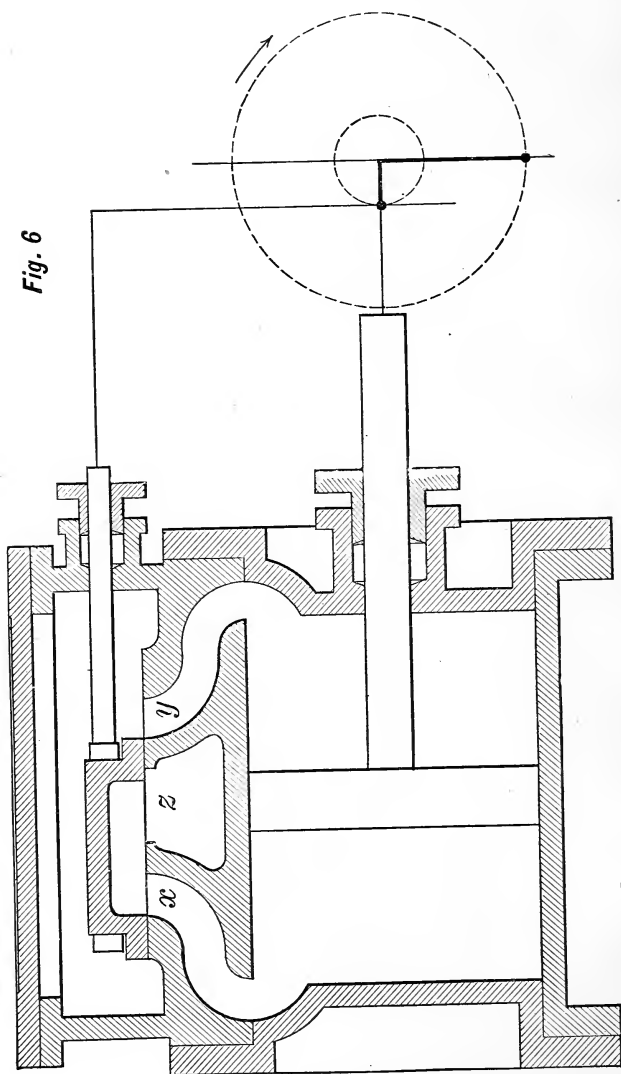
the correctness of this fact by supposing the eccentric so located, and then following the motion through a revolution.

Throughout this book, whether shown or not, it will be understood that the cylinder is located as in the figures already explained, i.e., to the left of the shaft; and unless otherwise specified, that the direction of rotation is the same as in these figures, i.e., "over."

DEFECTS OF THE PRIMITIVE ENGINE.

With the construction of Figs. 2–6 the opening and closing of the ports are coincident with the passing of the centre by the crank. Economy of steam and successful running require that the following changes be made in this distribution of the steam :

I. The opening of the steam port or "admission" should occur slightly before the crank reaches the centre.* In a general sense this is called giving the valve *steam lead* or simply *lead*. In a more strict sense, that term means the width of opening in fractions of an inch which the valve has given to the steam port at the instant the crank passes the centre.

II. The closing of the steam port or "cut-off" should occur a good deal before the crank passes the centre.

III. The opening of the exhaust port or "release,"

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* Of late years a difference in practice has arisen in this respect. Some makers now set their valves to open the port just as the crank passes the centre, and in some cases the admission is delayed until after that event. The statement in the text, however, represents general practice. This subject will be referred to again farther on.

Fig. 7
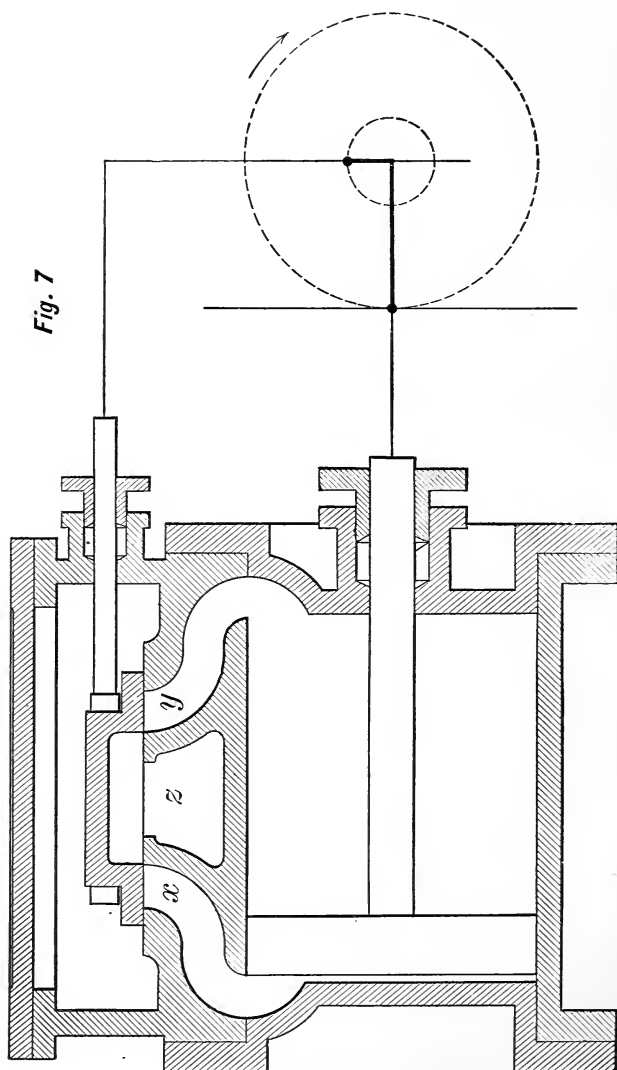
and its closing or "compression," should occur earlier
than the opening of the steam port, but not so early as
its closing. As the width of port opening to steam as
the crank passes the centre is called steam lead, so the
width of opening to the exhaust at the same instant is
called exhaust lead or inside lead (compare page 4).

These changes in the steam distribution are brought
about by two changes in the valve gear: (I.) The valve
is given lap, and (II.) The eccentric is advanced on the
shaft ahead of the position given.

LAP.

Fig. 7 is a reproduction of Fig. 2, with the addition
of outside or steam lap to the valve. There is no
change in the exhaust side of the valve nor in the
angular position of the eccentric, and it is obvious that
the ports will be opened and closed to the exhaust as
the crank passes the centre exactly as before; but the
port x will not be opened to steam until the valve has
been carried to the right an amount equal to its lap.

This will happen as shown in Fig. 8, when the shaft
has turned through an angle def, such that df, or what
is the same thing, eg, is equal to the lap. This angle
def is called the *lap angle*. The valve opens port x to
steam when the edge of the valve passes the edge of
the port going to the right, and it closes it when the
edge of the valve passes the edge of the port going to
the left. The position of the valve is the same at clos-
ing as at opening, the only difference between the two
acts being in the direction of the valve's motion; con-

Fig. 8
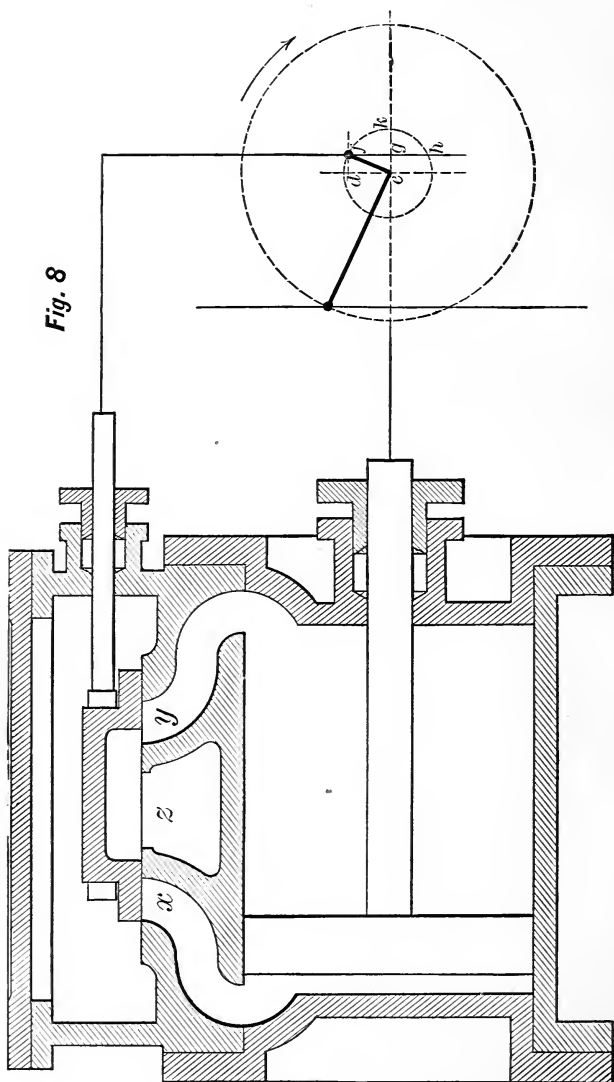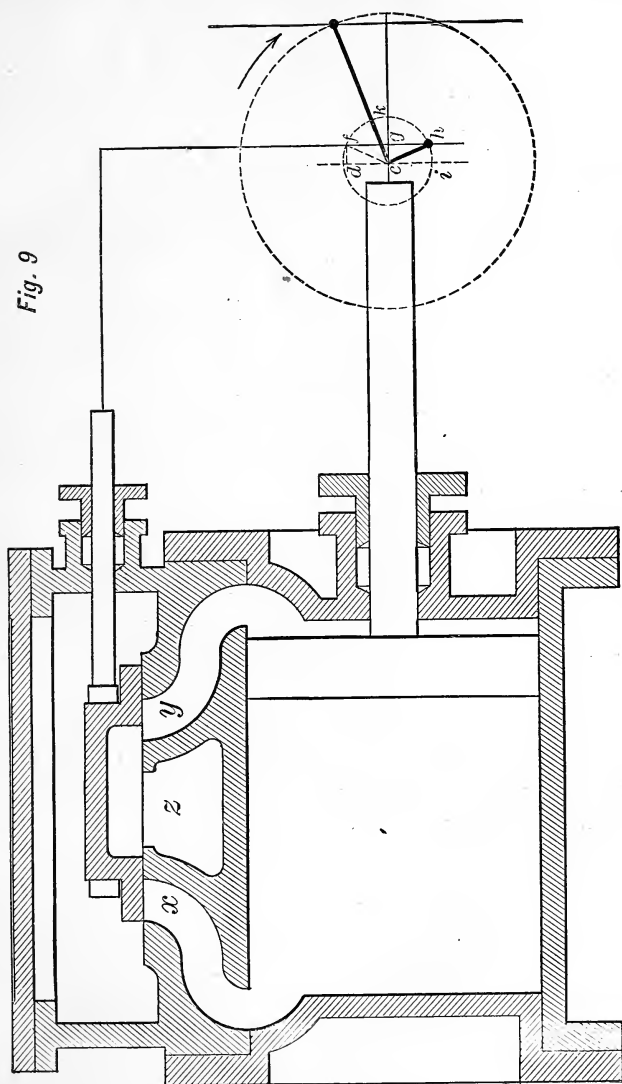
Fig. 9

sequently the port must close with the eccentric at *h*, vertically below *f*, as shown in Fig. 9. From this, two important and fundamental facts can be learned:

I. During the time that steam was being admitted the eccentric (and with it the shaft and crank) turned through the angle *feh*. Now *feh* is equal to a semicircle less *def* and less *hei*, that is, a *semicircle less twice the lap angle*, and this is the first result of the addition of lap to the steam side of the valve. During the admission of steam the crank turns through an angle equal to a *semicircle less twice the lap angle*.

II. In the case of the primitive valve of Figs. 2–6, the port began to open with the eccentric in the position of Fig. 2. It attained its greatest opening in the position of Fig. 4, and the maximum width of port opening was equal to the throw of the eccentric. With the valve of Figs. 7–9, however, the port does not begin to open until the eccentric reaches the point *f*, and the remaining travel *gk* only is available as port opening. This width of opening, *gk*, is equal to *ek* less *eg*, that is, to the *throw of the eccentric less the lap of the valve;* and it follows at once that if a valve with lap is to give the same port opening as another without lap, the eccentric throw of the former must exceed that of the latter, and the greater the lap the greater must be the throw.

ANGULAR ADVANCE.

The last section has shown how, by the addition of lap, the period of port opening may be shortened as desired. While, however, the method of regulating

the length of period of port opening was pointed out, nothing was said about properly timing the opening or closing of a valve having lap with reference to the position of the piston, and in point of fact in Fig. 8 the port opening to steam occurred long after the crank had passed the centre. The correct timing of the events of the stroke, and especially of the admission of steam, is obtained by advancing the eccentric around the shaft from the position thus far shown. In order to give admission to the steam at the instant the crank passes the centre, it is necessary to first locate the crank on the centre, then to advance the eccentric around the shaft such an amount as to draw the valve to the right a distance equal to the steam lap, and finally to secure the eccentric in that position. Such a setting of the eccentric is shown in Fig. 10, in which the eccentric has been turned forward until the distance *df* or its equal, *eg*, is equal to the lap. Such advance of the eccentric on the shaft is called the *angular advance* or the *advance angle* of the eccentric; and if the admission is to occur with the crank on the centre, as in this instance, the advance angle of Fig. 10 is equal to the lap angle of Fig. 8. Having secured a proper admission to the steam, it is proper to inquire next into the effect which this change in the angular position of the eccentric has had on the other events of the stroke. It is sufficiently obvious that turning the eccentric forward a given angle would simply cause each event to occur that much earlier in the rotation of the crank. After steam lap had been added, and before the advance of the eccentric, the cut-off occurred with the crank lacking one lap angle of having reached

Fig. 10
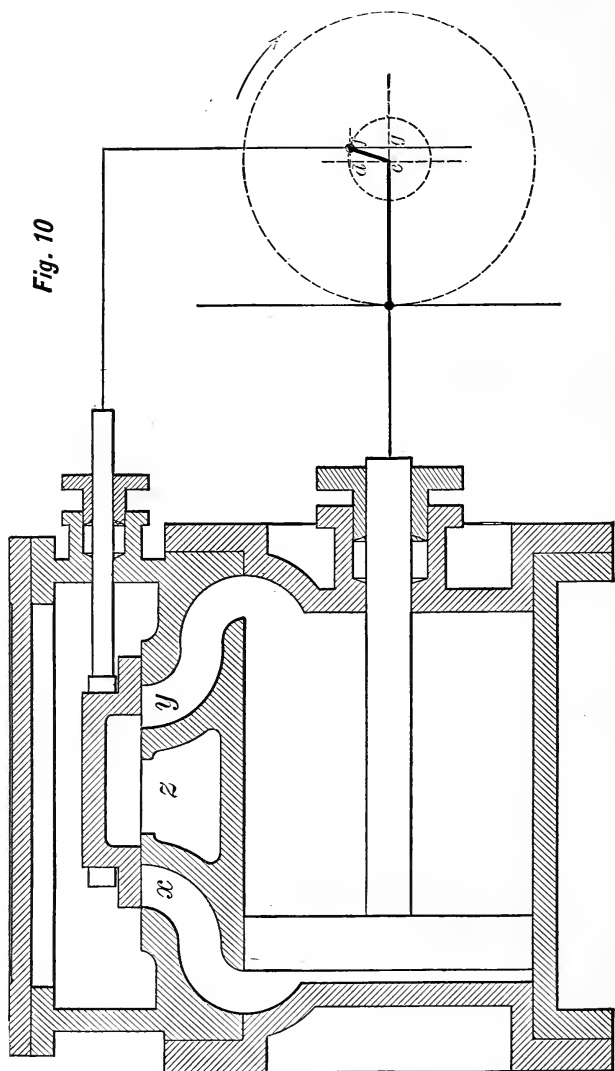
the centre (see Fig. 9). Advancing the eccentric one lap angle will cause cut-off to occur one lap angle earlier still, or with the crank lacking two lap angles of having reached the centre. Before the advance of the eccentric, the opening and closing of the ports to the exhaust occurred as the crank passed the centre (see Figs. 5 and 7). Advancing the eccentric one lap angle will therefore cause both release and compresssion to occur with the crank lacking one lap angle of having reached the centre.

Summarizing then, the valve has had steam lap added to it, and the eccentric has been advanced by an angle equal to the lap angle, and the resulting steam distribution is as follows: Admission occurs as the crank passes the centre; cut-off occurs two lap angles before the centre; and release and compression occur one lap angle before the centre.

Throughout this book the advance angle will be designated on the diagrams by the letter δ (delta).

LEAD.

It was explained on page 13 that the admission of steam should take place slightly before the crank reached the centre, and that such early admission was called lead. In the last section, for the sake of simplicity, the admission was supposed to occur as the crank passed the centre. In other words, the lead was made zero. It now becomes in order to examine the method for the introduction of lead, and the changes in the other events of the stroke which follow. In Fig.

Fig. 11
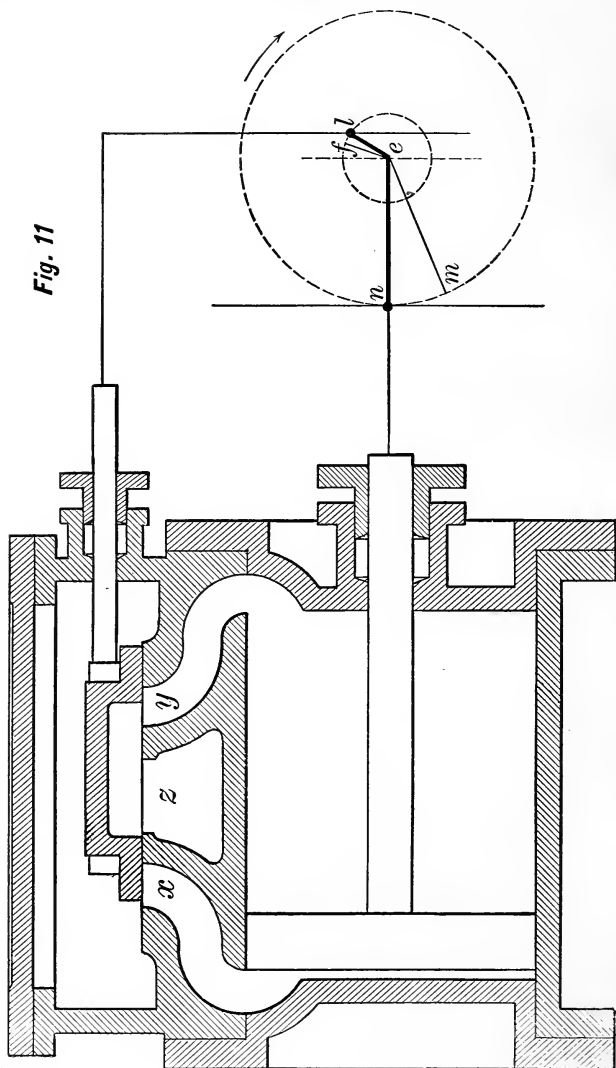
10 the eccentric was advanced to cause admission to occur on the centre. If it is proposed to give the valve lead, the eccentric must be advanced still further, so as to draw the valve to the right an additional amount equal to the lead desired. In Fig. 11 this additional advance has been made, the eccentric having been moved from f of Fig. 10 (reproduced in Fig. 11) to l. The angle *fel* is called the *lead angle*, and it follows that in all cases the angular advance is equal to the lap angle plus the lead angle. If the lead is zero, then, as before found, the angular advance is equal to the lap angle. If in Fig. 11 the crank shaft be turned backward, the valve will close the port when the eccentric reaches point f and the crank stands at m, the angle *nem* being equal to the angle *fel*. In other words, the lead angle is equal to the angular distance which the crank lacks of having reached the centre when admission occurs.

It was found on page 19 that advancing the eccentric on the shaft a given angle advanced all the events of the stroke correspondingly, and the resulting distribution of steam with no lead was summarized on page 21. If now the valve have lead so that the angular advance be greater than the lap angle, the steam distribution given on page 21 is changed as follows :

Admission occurs with the crank lacking the lead angle of having reached the centre.

Cut-off occurs with the crank lacking two lap angles and one lead angle of having reached the centre.

Release and compression occur with the crank lacking one lap angle and one lead angle of having reached the centre.

An application of the above principles is all that is

necessary for the analysis of the steam side of any existing plain slide valve, as will be seen from the following example :

An eccentric has a throw of $1\frac{5}{8}''$, the valve has one-
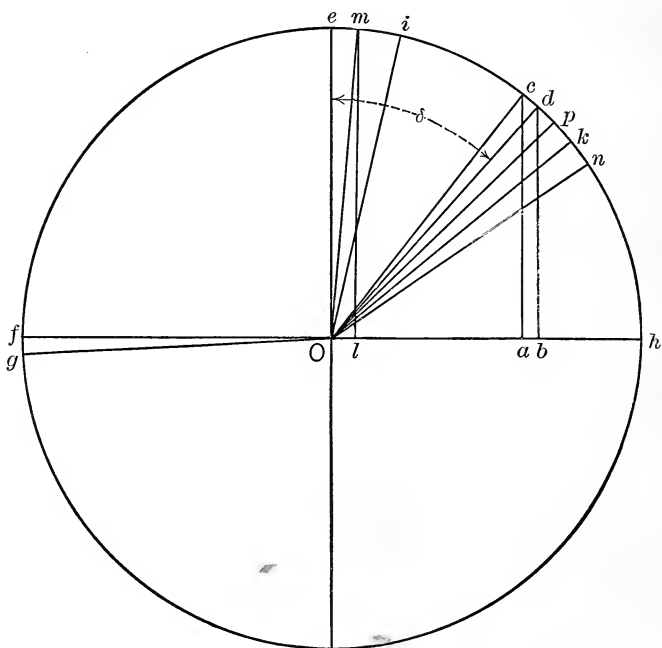
Fig. 12

inch steam lap, no exhaust lap, and the eccentric is set to give $\frac{1}{16}''$ lead. Required the greatest port opening, and the crank positions for admission, cut-off, release, and compression.

In Fig. 12 strike the circle with a radius equal to the

throw of the eccentric, and lay off *Oa* equal to the lap of the valve and *a b* equal to the lead. Erect perpendiculars from points *a, b*, giving points *c, d*. Now *eOc* is the lap angle, *cOd* is the lead angle, and *eOd* is the angular advance. Lay off *fg* equal to *cd*, giving *Og* the crank position for admission. Make *hi* equal to twice *ec* plus *cd*, giving the crank position *Oi* for cut-off. Make *kh* equal to *ec* plus *cd*, giving *Ok* the crank position for release and compression. Finally, *ah* is the greatest steam port opening.

The student should familiarize himself with the principles thus far treated, by the solution of a variety of problems similar to the following :

Problem I. Eccentric throw $1\frac{3}{8}$, lap $\frac{3}{4}$, lead $\frac{1}{16}$. Required crank positions for lead, cut-off, compression and release, and port opening to steam.

Problem II. Eccentric throw $1\frac{3}{4}$, lap $1\frac{3}{16}$, lead $\frac{3}{32}$. Required as before.

EXHAUST LAP.

The addition of lap to the inside of the valve has the same effect upon the opening and closing of the port to the exhaust that steam lap has upon opening and closing it to steam, i.e., it delays the opening and hastens the closing, and it delays the opening and hastens the closing by an angle of rotation of the crank equal to the exhaust lap angle—that angle being found exactly as was the steam lap angle in Figs. 8 and 12. Returning to Fig. 12, if the valve were to have an addition of $\frac{1}{8}$ inch inside lap, its effect upon the release

and compression could be found as follows : Make *Ol* equal the inside lap, and by the line *lm* find the exhaust lap angle *eOm*. From *k* lay off *em* both upward and downward. Draw the radii *On* and *Op*, and the results are *On* for the crank position at release, and *Op* for compression. It is unnecessary to go in detail through the results following the introduction of negative inside lap. It will be seen at once that its results are the direct reverse of the preceding, i.e., to hasten the opening and delay the closing of the ports. If in the illustration preceding, the valve had been given $\frac{1}{8}$ negative inside lap, the crank positions found for release and compression would have changed places.

It was shown on page 18 that the steam port opening given by a valve having steam lap is equal to the throw of the eccentric less the steam lap, and it follows by the same course of reasoning that the exhaust opening is equal to the throw less the exhaust lap. Since the exhaust lap is always less than the steam lap, it follows that the port opening to exhaust is always greater than to steam.

BACKWARD ROTATION.

It was explained on page 11 that with a primitive valve the eccentric location for rotation in the reverse direction would be diametrically opposite that shown in Figs. 2–6. For a valve having lap, the position for reverse rotation is found by laying off the advance angle in the direction of the proposed rotation from the position for a primitive valve. The effect of a rock

shaft in the valve motion (for an example of which see any American locomotive) is to reverse the motion of the valve as compared with the eccentric, and hence to require a location of the eccentric which will provide for this reversal. The position of the eccentric for either direction of rotation, and with or without a rocker, may be located from the following facts :

I. Without a rocker the eccentric for a primitive valve is 90° in advance of the crank in the direction of the rotation. With a rocker the eccentric is behind the crank.

II. The advance angle is laid off in all cases in the direction of the rotation from the position for a primitive valve.

One qualification should be added to the above, as follows : In all the cases thus far shown, the location of the eccentric for the primitive valve is as stated at right angles to the crank. In certain cases, owing to the character of the connections between the eccentric and valve, this is not true. For an example see Figs. 58 and 59. In cases of this kind the location of the eccentric can be found as follows : Carry the centre of the eccentric strap to the centre of the crank shaft. Through the centre of the shaft draw a line perpendicular to the location of the eccentric rod thus found. This perpendicular gives the location of the eccentric for the primitive valve, and the angular advance is to be laid off from it in the direction of the rotation.

THE BILGRAM DIAGRAM.

Any existing slide valve can be analyzed by the methods that have been followed in explaining the action of the valve, and new valves could be designed by a tentative application of the same methods. Such a plan of procedure, however, would be exceedingly tedious, and much ingenuity has been expended in devising briefer and better methods. Of these, by far the best is the diagram devised by Mr. Hugo Bilgram, and explained below. The chief office of such a diagram is to show briefly and accurately the position of the valve for any and every position of the crank.
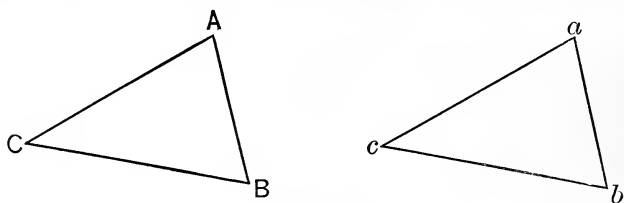
Fig. 13

The demonstration of the Bilgram diagram depends upon the following theorem of geometry: In Fig. 13 let *ABC* and *abc* be two triangles, such that any two of their angles, as those at *A*, *C* and *a*, *c*, and any one side, as *BC* and *bc*, are respectively equal. Then this theorem asserts that all of the other parts of the triangles are equal, i.e., angle *B* to *b*, side *AC* to *ac*, and side *AB* to *ab*.

In Fig. 14 let *A* be the dead-point location of the crank, and *B* be the corresponding position of the ec-

centric centre, δ being the angle of advance. It is obvious enough that the valve is now located a distance Bb (equal to the sum of the lap and lead) to the right of its middle position. Imagine the crank to turn through the angle α to a new position A'. The eccentric will turn through an equal angle α to its new
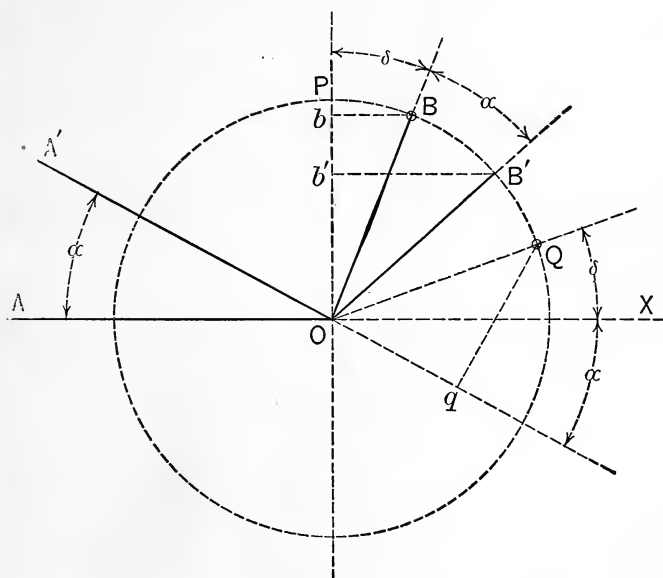

Fig. 14

position B', and the valve will then be located a distance $B'b'$ to the right of its middle position. Lay off the angle δ upward from OX, and thus locate a fixed point, Q. From Q drop Qq perpendicular to the new crank position extended. There are thus formed two triangles, $B'b'O$ and QqO, and in them $B'O$ equals

QO, since both are radii of the same circle. Angles $B'b'O$, QqO are equal, because both are right angles; and finally, angle $B'Ob'$ equals QOq, since each is equal to δ plus α. The two triangles have thus two angles and a side of one, respectively equal to two angles and a side of the other, and it follows that the triangles are equal in all their parts, and hence Qq equals $B'b'$. $B'b'$ is the distance which the valve has travelled from its central position for crank position A', and it hence follows that Qq likewise equals that distance.   The same demonstration can be made for any other crank position as well as for A', and the following general fact is thus established : Lay off the advance angle above the centre line, and thus locate the fixed point Q.   Draw any crank position desired, and extend it if necessary.   From the fixed point Q drop a perpendicular to the crank line, and the length of the perpendicular will be equal to the distance of the valve from its central position for the crank position taken.   For rotation in the reverse direction, points B and Q would fall below instead of above the centre line.

The length of the perpendicular Qq gives the distance of the valve from its central position, but it does not of itself show whether the valve is located to the right or to the left of its middle position.   That fact will be determined instinctively after a little practice in the use of the diagram ; but if desired it can be determined by the following consideration : Referring to Fig. 14, that side of the crank and of its imaginary extension facing the space toward which the crank is revolving may be called the face side of the crank, and the opposite side may be called the rear side.   If the

perpendicular Qq falls upon the face side of the crank, the valve is to the right of its middle position; if the perpendicular falls upon the rear side, the valve is to the left of its middle position.* As has been explained, the greatest port opening is equal to the throw of the eccentric OQ, Fig. 14, less the lap; and it is also true that for any position of the crank, the port opening which exists at that position is equal to the displacement of the valve from its central position less the lap, i.e., to the value of Qq for that position, less the lap. In other words, if for any crank position the value of Qq be found, and from it the steam lap be taken, the result will be the distance which the steam port stands open for that crank position. If, on the other hand, the exhaust lap be taken from it, the result will be the distance which the exhaust port stands open. This subtraction can be conveniently made by striking two circles L, l from Q as a centre, and with radii equal to the steam and exhaust laps respectively, as is done in Fig. 15. In case the inside lap is negative, it of course increases instead of decreases the port opening to exhaust. Throughout this book, positive lap will be shown by full circles, and negative lap by dotted circles.

Starting with the position A, Fig. 15, the length of the perpendicular which locates the valve is Qq, and the width of opening of the port to steam is aq; A being the dead-point position of the crank, aq is the lead of the valve. Similarly, bq is the exhaust lead. In Fig. 16 the valve is shown in position for crank position A.

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* This takes it for granted, as explained on page 13, that the position of the cylinder is to the left of the shaft.

The opening of the port c to steam is equal to aq of Fig. 15, and the opening of port d to exhaust is equal to bq. Similarly, the displacement e of the valve from its centre is equal to Qq.* As the crank revolves, Qq gradually lengthens until the crank reaches position B perpendicular to OQ, when Qq becomes QO, which is its

Fig. 15

greatest value. The valve now stands at its extreme right hand position as shown in Fig. 17, the ports being open to their greatest amount—the steam port by a

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* It will be understood that the distances stated as equal are not so shown in the cuts, as they are necessarily drawn to different scales.

width $a'O$, and the exhaust port $b'O$. Passing B, the valve returns towards its central position, and at C the
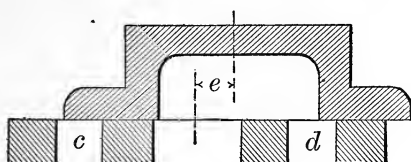
Fig. 16

displacement has been reduced to equality with the steam lap. The port c is therefore closed to steam, and
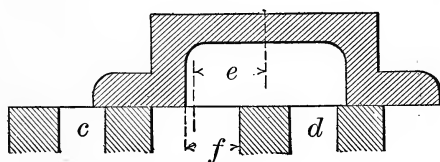
Fig. 17

cut-off takes place as shown in Fig. 18. At D, port d is closed to the exhaust, Fig. 19, and compression be-
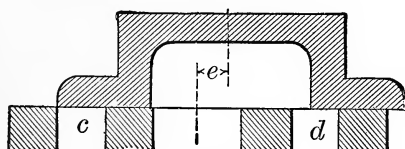
Fig. 18

gins. At F, port c is opened to exhaust, Fig. 20, and release occurs. At G the valve is ready to open port d for the return stroke, Fig. 21 ; and at H the valve has

opened port d by the amount of the lead. The positions of the crank for the return stroke are readily found
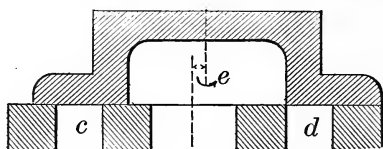
Fig. 19

by extending the crank lines beyond the centre. Thus I is the lead position, K the release, M the compression,
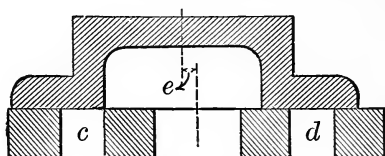
Fig. 20

and N the cut-off. Had there been no exhaust lap, release and compression would have occurred simultane-
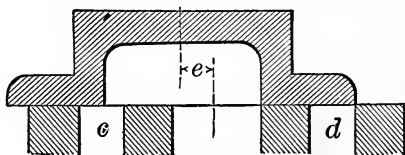
Fig. 21

ously at E; and had the exhaust lap been negative, release and compression would have exchanged places, and the maximum opening to exhaust would have been Ob''.

Consideration of this diagram will recall and enforce the essential effect of lap, as stated on page 18 ; i.e., to shorten the period during which the port is open. Thus with steam lap the steam port is open when the crank is moving from I to C and from G to N, and with exhaust lap the exhaust port is open from K to D and from F to M. With negative inside lap, on the other hand, the port is open during more than half a revolution, i.e., from M to F.

The principles laid down should be fixed in the mind by the solution of practical problems similar to the following :

Problem III. Throw of eccentric $2''$, steam lap $1\frac{1}{8}$, exhaust lap $\frac{1}{4}$, lead $\frac{1}{8}$. Required port opening and points of cut-off, release, and compression.

Problem IV. Travel of valve $3''$, steam lap $\frac{1}{2}$, negative exhaust lap $\frac{3}{16}$, lead $\frac{1}{8}$. Required as in the last problem.

This diagram is of use not only in analyzing existing valve motions as in the preceding problems, but also in designing new ones to meet required conditions. The method of using it for this purpose is best shown by an illustrative example, as follows :

The valve for a certain engine is to have a steam-port opening of $\frac{5}{8}''$, a lead of $\frac{1}{16}$; is to cut off the steam at $\frac{3}{4}$ of the stroke, and open the exhaust at 95 per cent of the stroke. Required the inside and outside lap, the throw and advance angle of the eccentric, and the point of exhaust closure.

In Fig. 22 make AB equal to the length of the stroke, using a scale of three inches to the foot. Make Aa

equal $\frac{3}{4}$ of *AB*, and *Ab* equal .95 of *AB*. Draw the
semicircle $A'a'b'B'$ to represent the path of the crank,
and project to it the points *a*, *b*. Draw *Oa'* and *Ob'*,
which are the crank positions for cut-off and release.
Draw *cd* such that *de* is equal to the lead opening, $\frac{1}{16}$
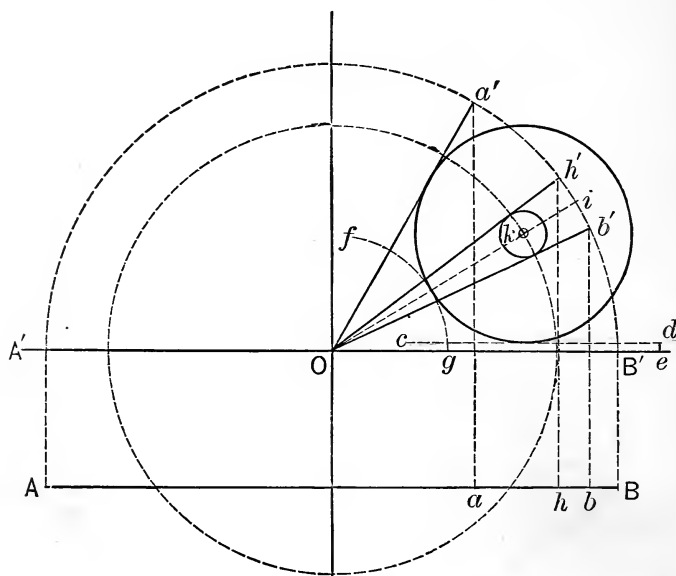
Fig. 22

inch; and strike the arc *fg* with radius equal to the
port opening. Find by trial the centre and radius of
the steam lap circle such that it shall be tangent to *Oa'*,
cd, and *fg*. From the same centre strike the exhaust
lap circle tangent to *Ob'*. Draw *Oh'* tangent to the ex-
haust lap circle and project *h'* to the stroke line, giving *h*.

Measuring the diagram, the results sought are, outside
lap $\frac{9}{16}$ inch, inside lap $\frac{1}{8}$ inch, throw of eccentric $1\frac{3}{16}$
inch, advance-angle iOB', point of exhaust closure h,
which is 90 per cent of the stroke. It may be observed
further, that the exhaust port opening is Ok; in this
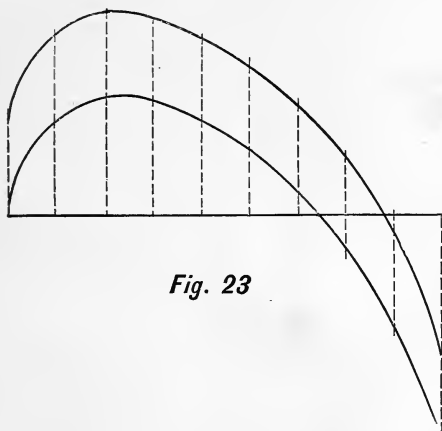
Fig. 23

case and in all others this is more than sufficient for
the purpose, and hence no particular care is necessary
in relation to it.

An interesting and profitable exercise in this connec-
tion is to make a diagram showing by a continuous line
the varying width of port opening throughout the
stroke. This is illustrated in Fig. 23, in which the con-
tinuous base line represents the stroke of the piston of
Fig. 22 divided into tenths. At each division a per-
pendicular is erected, and on this perpendicular is laid
off the opening of the ports to steam and exhaust for
that position of the piston—this opening being obtained
from Fig. 22. Through the points thus found the

curved lines are drawn—the upper one for the exhaust
port and the lower one for the steam port. The cross-
ing of the base line by the curved lines shows the points
of cutting off and compression, respectively; and the
extension of the curved lines below the base line shows
the distances by which the edges of the valve have
closed the ports. This diagram shows at a glance how
gradual is the cutting off of the steam. Such diagrams
are exceedingly useful in connection with the study of
independent cut-off valves, many of which will not give
flattering results when subjected to this analysis.

LAYING OUT THE SLIDE VALVE.

The diagram Fig. 22 gives all the dimensions neces-
sary for laying out its valve, except the width of the ex-
haust cavity, and that is determined at once by draw-
ing the valve and its seat with the valve at one extreme
of its travel, that is, in the position already shown in
Fig. 17. Referring to Fig. 1, it is clear that the width
of the acting face of the valve is equal to the outside
lap *plus* the width of the port *plus* the inside lap. In
order to determine all the dimensions of the valve face
and seat proceed as follows : Lay down the width of
the left hand port (rules for which will be given farther
on) and the width of the bridge (usually made equal to
the thickness of the cylinder). On these locate the
acting face of the valve with the port open to steam to
the greatest amount intended. From the exhaust
edge of the acting face lay off the distance f, Fig. 17, to
equal or slightly exceed the width of the port, thus com-
pletely determining the exhaust cavity in the cylinder.

From the right hand edge of this cavity the remaining bridge and port are to be laid off the same as the left hand side. The valve seat being completed, and the steam and exhaust laps being known, it is easy to complete the drawing of the valve. The proper determination of the distance f, Fig. 17, as above, is all that need be considered in designing the exhaust cavity in the cylinder. If this cavity be made too narrow, it will cramp the exhaust ; if too wide, it will add unnecessarily to the size of the valve and to the steam pressure upon it, and hence to the friction and wear and tear on all the valve gear. Further than this, the size of the exhaust cavity has no influence on the valve motion.

It will be observed that in the valve diagram Fig. 22, the lines for the piston and crank circle are drawn to a reduced scale, but the lines for the valve and eccentric are full size. The reduced scale for the crank dimensions is for convenience. The valve dimensions should always be made full size.

* VELOCITY OF THE VALVE.

Referring to Fig. 14, it is obvious that the valve will move with its greatest velocity when the eccentric is at P. At this point its velocity may be represented by the eccentric throw OP. At any other position of the eccentric as B, the valve will move with a velocity proportional to the leverage with which the eccentric acts upon it, that is, Ob. Similarly at B' the velocity of the valve will be represented by Ob'. In the original demonstration of this diagram it was shown that the triangles $B'b'O$ and QqO are equal in all their parts. It

hence follows that Oq equals Ob'. In other words, if a
perpendicular be drawn from the point Q to any crank
line, the distance from the centre of the shaft to the
foot of that perpendicular will represent the velocity
with which the valve is moving with the crank in that
position. Quick closure of the port in cutting off steam
is considered a merit in a valve motion, and this prop-
erty of the Bilgram diagram furnishes a ready means
of comparing the merits of different valve gears in this
respect. In Fig. 15 the perpendicular Qc will deter-
mine the distance Oc, which represents the velocity
with which the valve is moving at the instant of
cutting off.

LIMITATIONS OF THE PLAIN SLIDE VALVE.

Careful study of the Bilgram diagram will explain
the features of the common slide valve which have usu-
ally been considered to limit its application to cases
where a comparatively late cut-off was to be employed.
Thus, in the example of Fig. 22, let the given condi-
tions be the same, except that cut-off is to be at half
stroke instead of three quarters, and let there be no
inside lap. The results are shown in Fig. 24, where
it will be seen that the throw has increased to two
inches and the steam lap to one and three eighths
inches, while the common point of compression and re-
lease has gone back to bh—85 per cent of the stroke.
At still earlier points of cut-off these features become
still more marked, the travel of the valve rapidly in-
creasing and the release and compression becoming
more and more premature. The increased lap and
travel increase directly the size and duty which the

parts have to perform. Further, as will be seen by referring to the section on laying out the slide valve, and to Fig. 17, they increase the size of the exhaust cavity, and so add to the size of the valve and to the steam pressure upon it. The release can be made later

Fig. 24

by the addition of exhaust lap, but this involves a still earlier compression. From these considerations it has been generally held and taught that the plain slide valve could not be profitably employed for cut-offs shorter than one half or five eighths stroke. The

methods which have been adopted to overcome the above difficulties will form the subject of a later chapter,

THE AREAS OF THE PORTS AND PIPES.

It will be seen from the foregoing that the width of port opening is an essential factor in the design of a valve motion. The exact meaning of the term port opening in this connection should be clearly understood. By that term is to be understood the extreme distance of the steam edge of the valve from the steam edge of the port. This distance may, and often does, exceed the width of the port—that is, the valve may have over-travel to secure certain real or fancied advantages. In engines with fixed eccentric, which are now under consideration, the only benefit of such over-travel is to increase the sharpness of the cutting off. This, in the author's opinion, is not worth its cost, and hence he does not practise nor recommend it. In locomotives and shifting eccentric engines the travel of the valve is shortened at the early cut-offs, and in such engines, in order to secure sufficient port opening at the early cut-offs, it is proper and necessary to give over-travel at the late ones.

It is clear that the area of the port opening should have a proper relation to the size and speed of the engine. It will be furthermore clear without extended explanation, that in engines having separate admission and exhaust ports (for example, the Corliss) the exhaust passage should have a greater area than the admission passage. In engines using the same passage for both purposes, to which this book relates, that passage should be proportioned to meet the requirements of

the exhaust, and then, if desired, it need not be opened to steam any wider than is necessary for its use as a steam port.

In determining the area of a pipe or passage it is treated as though the velocity of the steam through it were equal to the velocity of the piston multiplied by the ratio of the area of the piston to the area of the port or pipe. Of course, owing to the expansion of the steam at release, this is not strictly true, but it simplifies the determination of the dimensions, and as long as the rules and tables make allowance for its lack of precision, the results are the same as though a more complicated process were gone through.

As the result of experience and experiments, the proper velocities of the steam through the various passageways are as follows:

Through the steam pipe 8000 feet per minute.

Through the exhaust port 6000 feet per minute.

Through the exhaust pipe 4000 feet per minute.

For free admission of steam the port should be opened three fourths of its width. From the above data the following table is constructed for convenient use:

Piston Speed, Feet per Minute.	Diameter of Steam Pipe (Diameter of Piston = 1).	Diameter of Exhaust Pipe (Diameter of Piston = 1).	Area of Exhaust Passage (Area of Piston = 1).
200	.158	.223	.033
250	.176	.248	.042
300	.194	.272	.050
350	.209	.294	.058
400	.224	.314	.067
450	.237	.333	.075
500	.250	.353	.083
550	.260	.368	.092
600	.274	.385	.100

The table determines the diameters of the steam and exhaust pipes at once, but it gives the area only of the port, leaving its length and breadth to be determined by the designer. The practice in this particular is very diverse. In shifting eccentric automatic engines, which form the subject of Part II, and in which every expedient must be employed to secure sufficient port opening, the length of the ports is often made to equal or even exceed the diameter of the cylinder; but in plain slide valve engines of the usual type, a length of about three quarters the cylinder diameter more nearly represents average practice. This length determined, it is only necessary to divide the area of the passage by it to determine the width of the port, and three fourths of this will give the port opening to be used in laying out the diagram.

A " rule of thumb" which is in very common use is to make the steam pipe one fourth the diameter of the cylinder, and the exhaust pipe one third. At slow speeds this rule gives an excess of capacity over the requirements, to which of course there is no objection; but at high speeds it gives a deficiency. On high grade engines, where the best results are sought, steam pipes are seen as large as one third and exhaust pipes one half the cylinder diameter.

All the principles thus far given will be found required in the solution of the following

Problem V.—An engine with a $10'' \times 15''$ cylinder is to run at 200 revolutions per minute. Cut-off is to be at $\frac{5}{8}$ stroke, release at .93 stroke, and lead is to be $\frac{3}{32}''$. Required the diameters of steam and exhaust pipes, the

dimensions of the ports, the travel, and the steam and exhaust laps of the valve.

* THE ANGULAR VIBRATION OF THE CONNECTING ROD.

As has been explained, the slotted cross-head was adopted in the preceding to avoid certain distortions which are incident to the use of the connecting rod. It is now proper to discuss these distortions, and explain the methods for neutralizing their effects.

With the slotted cross-head the position of the piston
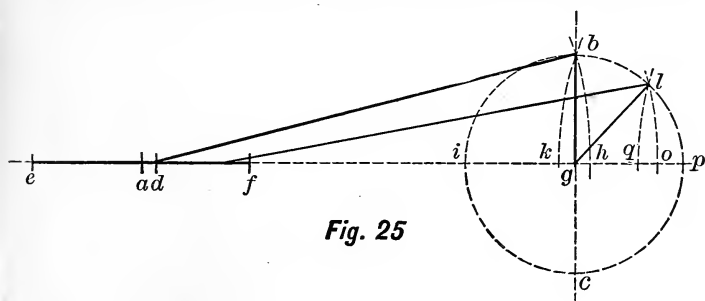
Fig. 25

or cross-head in its stroke for any crank position is found by simply projecting the crank position to its horizontal diameter, or a line parallel thereto, by means of a straight projecting line, as was done in Figs. 12, 22, and 24. Fig. 25 is a skeleton diagram of the usual connecting rod and crank. It is obvious that if the crank pin end of the connecting rod be disconnected from the crank pin and carried to the centre of the crank shaft the cross-head pin will occupy its central position *a*. If from this position the crank pin end be carried to either "quarter" position of the crank pin *b*, *c*, the cross-head

pin will be drawn toward the shaft and will occupy the
position *d*. For the forward stroke the position of the
cross-head is measured from *e* as a starting point, and
hence the cross-head and piston have moved *too far*
by the distance *ad*. For the return stroke the position
is measured from *f* as a starting point, and hence the
cross-head and piston have *not moved far enough* by
the same distance. If the valve motion were laid out
by the preceding methods to cut off steam at half stroke,
it would in fact cut off later than half stroke for the
forward stroke and earlier for the return. The same
distortion takes place at all other positions of the crank
except at the centres, though to a less degree ; and it
follows that all the events of the stroke except the lead
occur too late in the forward stroke and too early in
the return. The amount of this distortion can be found
for any position of the parts, as is done in Fig. 25, for
the position shown, by striking an arc with radius equal
to the length of the connecting rod, the distance *gh* be-
ing equal to *ad*. Striking this arc is, in fact, projecting
the point *b* to the centre line with the circular arc in-
stead of a straight line, as has heretofore been done ;
and in order to find the true relation between the posi-
tions of the piston and crank, it is only necessary to
project the one to the other by means of such circular
arcs with radius equal to the length of the connecting
rod. It is often convenient to measure the piston posi-
tions for the forward and return strokes from the same
starting point as *i*, and in order to do this it is only
necessary to strike the arcs for the forward and return
strokes from opposite sides of the shaft. Thus, with
the crank on the quarter, the piston will have moved

through the distance *ih* for the forward stroke and *ik* for the return. Similarly, if the crank move through an angle *ibl* for the forward stroke, the piston will have moved through the distance *io* ; and if the crank move through the same angle from *p* on the return stroke the piston will have moved through the distance *iq*. The directions of these distortions for the two strokes are best distinguished by remembering that the effect of the connecting rod is always to draw the piston too near the crank.

The amount of these distortions will diminish if the length of the connecting rod be increased, and if a connecting rod of infinite length be conceived, the distortions will disappear. Hence a piston motion without distortion, such as is given by the slotted cross-head, is often called the motion due to a connecting rod of infinite length.

Since the speed of the crank's rotation is uniform, and the piston must travel farther for a given angle of crank rotation in the forward than in the return stroke, it follows that the speed of the piston's motion is greater in the forward than in the return stroke.

These principles can be applied to the problems already given, and thereby determine the actual positions at which the various events occur. Fig. 26 is a reproduction of Fig. 22, but with the projections made by circular arcs instead of straight lines. It thus appears that with the valve there designed, the cut-off, instead of taking place as intended in Fig. 22, will really take place after a piston travel *Aa'*, Fig. 26, in the forward stroke and *Aa''* in the return. Similarly, the compression will take place after travels *Ah'* and *Ah''*, and the release after *Ab'* and *Ab''*. If preferred, the construc-

tion may be made by repeating the lap circles below
the centre line in position for the return stroke, as is
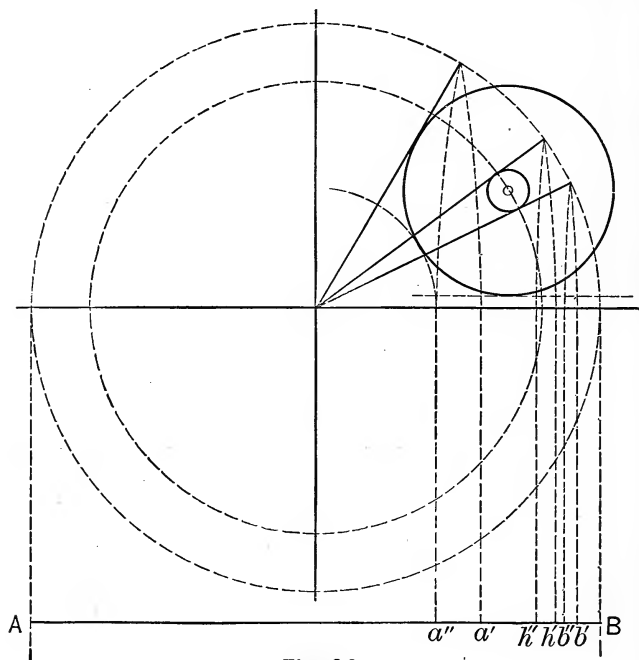
Fig. 26

done in Fig. 27. With this construction the measure-
ments are necessarily made from *A* for the forward
stroke and *B* for the return. Of these two plans that
of Fig. 26 possesses the advantage that it shows at a
glance the difference between the points of cut-off, etc.,
in the two strokes.

Problem VI. It is required to find the true positions
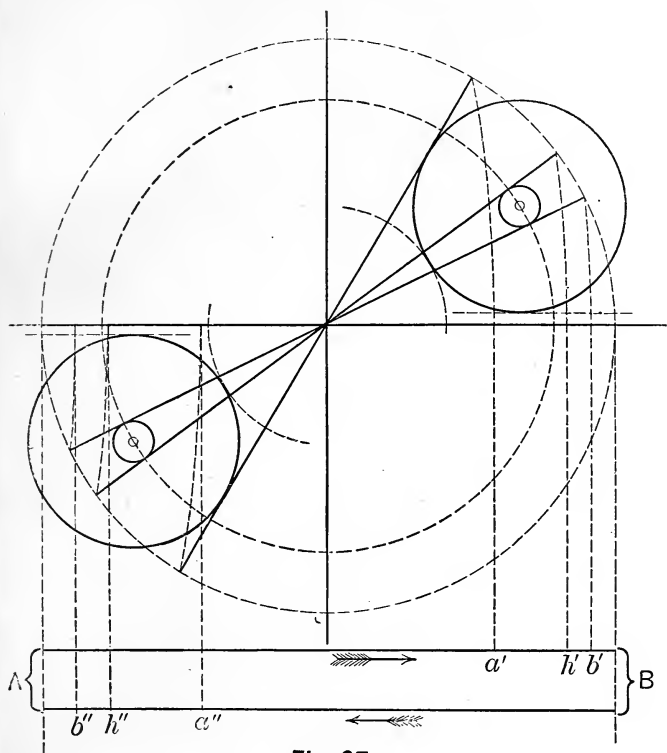for cut-off, release, and compression of the valve of

Fig. 27

Problem III. Length of connecting rod five times the crank.

* THE ANGULAR VIBRATION OF THE ECCENTRIC ROD.

As has been explained, the eccentric is in effect a crank and the distortions introduced by the connecting rod into the motion of the piston are likewise introduced by the eccentric rod into the motion of the valve.

In other words, if the distortions are not corrected, the
valve, like the piston, will always be too near the crank.
The throw of the eccentric is much less than the arm
of the crank, and the eccentric rod is proportionately
longer than the connecting rod ; hence the distortions
in the positions of the valve are absolutely and relatively
smaller than those in the positions of the piston.   Since
the effect of these distortions is to draw the valve too
near the crank, it follows that if they are not provided
for, the lead of the valve at the back end † of the cylinder
will be increased and for the front end diminished.   The
greatest port openings are measured with the eccentric
on the centre line of the engine, when these distortions
vanish ; and hence the two port openings will be equal.
It is important that the lead openings be equal, while
it is not particularly important that the maximum port
openings be equal, provided the smaller one be large
enough.   Hence in practice the eccentric rod or valve
rod is slightly lengthened to give equal lead at the two
ends, and the result is that the port opening is slightly
diminished for the forward stroke and slightly increased
for the return.   This lengthening of the eccentric rod
is effected in setting the valve for equal lead, and it
practically corrects the effects of the angular vibration
of the eccentric rod.

* EQUALIZED EXHAUST.

As has been explained, the setting of the valve for
equal lead practically neutralizes the effect of the an-

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† With apologies to the locomotive fraternity, the end of the cylinder
farthest from the crank will be called the back end.

gularity of the eccentric rod. Nothing, however, has
yet been done toward correcting the irregularities due
to the connecting rod, and that is the next subject to
be discussed.

If the valve has no inside lap, compression and re-
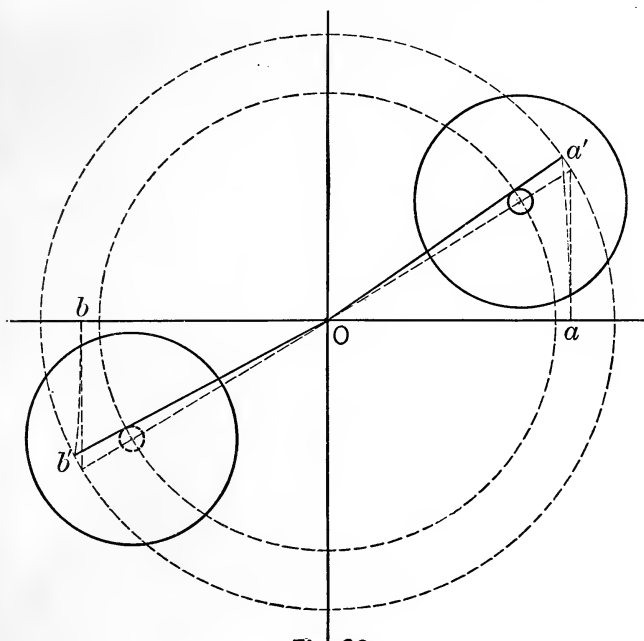
Fig. 28

lease are coincident, and the correction of one will like-
wise correct the other. It is desired to cause both these
events to occur earlier in the forward stroke and later
in the return stroke, and to accomplish this, it is only
necessary to give an appropriate exhaust lap to the end
of the valve nearest the crank shaft, and an equal nega-

tive exhaust lap to the other end.   In applying this cor-
rection, it should be remembered that the Bilgram dia-
gram gives the true relation between the positions of
the crank and eccentric, and that the distortions under
discussion are given to the piston through the connect-
ing rod. In Fig. 28 it is proposed to correct the release
and compression of the valve shown, which, in the first
instance, has no exhaust lap.   By the vertical projec-
tion lines the points of release a, b are found in the usual
way, and by means of the curved projection lines points
a', b' are found for the correct crank positions correspond-
ing to piston positions a, b.   If the release and compres-
sion are to take place at piston positions a, b, they must
take place at the crank positions a', b'.   Drawing the
crank lines $a'O$ and $b'O$, it is easy to add the exhaust lap
circles shown, from which measurement shows that for
that edge which effects exhaust at a' a positive lap of
$\frac{1}{16}''$ is required, and for b' a negative lap of the same
amount.   The resulting valve is shown in Fig. 29.   Had
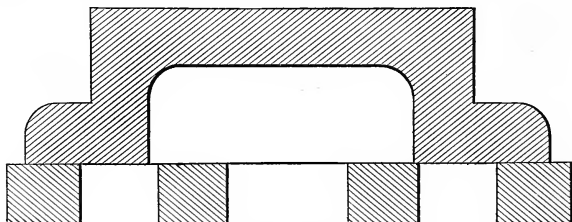
Fig. 29

the valve originally possessed exhaust lap, as in Fig.
30, exact equalization would have been impossible, al-
though a result could have been reached sufficiently near-
ly correct for all practical purposes. The piston positions

for compression a, b and release c, d are projected to the crank circle by circular arcs, as shown, giving the corresponding crank positions a', b', c', d'. It is apparent at once that the change of lap to give compression at a' is greater than the change to give release at d', and simi-
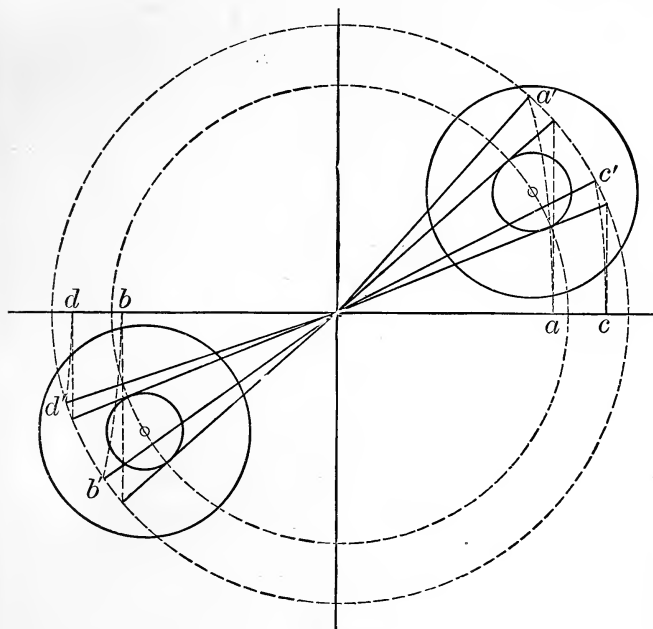
Fig. 30

larly for c' and b'. In such a case the best that can be done is to divide the difference, making the alteration in the lap half way between that called for by a' and d' for their end of the valve, and half way between that called for by c' and b' for their end of the valve. This subject will be returned to at the close of the next section.

* EQUALIZED CUT-OFF.

It was shown in the last section that by introducing inequality in the inside laps, the inequality of release or compression could be equalized—a change in one event being, however, accompanied by a change in the other. It is obviously possible to equalize the cut-off in a similar manner by making the outside laps unequal. As a change in the inside laps involved both release and compression, so will a change in the outside laps involve both admission and cut-off ; and since the valve, as thus far described, gives equal lead at the two ends of the cylinder, it follows that increasing one lap and decreasing the other would result in an unequal lead—in other words, cut-off equalized by such a method would involve unequal lead. Such a method is usually explained in detail in books of this character. Equality of lead is, however, of more importance than equality of cut-off, and hence the method is of no practical importance, and is not introduced here.

The following method † secures equality of cut-off without affecting the equality of the lead. It has no objectionable features, and is of general utility. Throughout the discussion, one fundamental fact must be kept in mind, viz.: The acts of opening and closing a port by a slide valve differ only in the direction of motion of the valve. The port is opened or closed, as the case may be, by the edge of the valve passing the

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† First published by the author in the *American Machinist* for March 14, 1889. It was invented independently by Professor Sweet and the author,—first, however, by the Professor.

edge of the port, and the position of the valve when cut-off takes place is the same as when admission takes place. Since the valve is mechanically connected to the eccentric rod pin, it follows that the position of that pin must be the same at cut-off as at admission.

Let it be proposed to design a slide valve to cut off steam at half stroke, with equal lead and cut-off.
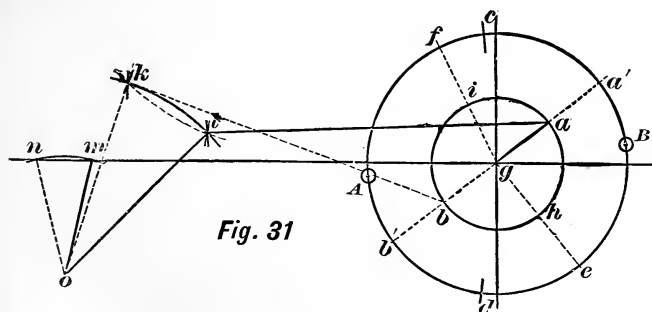
Fig. 31

First design the valve by the methods already explained, then strike the crank and eccentric circles of Fig. 31.* Locate points A, B, the positions of the crank pin for admission of steam, and the corresponding positions a, b for the eccentric centre. With radius equal to the length of the connecting rod, and with

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* In this and the following diagrams the throw of the eccentric is made disproportionately large, and the eccentric rods disproportionately short, to add to the clearness of the constructions without unnecessarily large diagrams. This gives the appearance of a distorted valve movement; but with working proportions these apparent distortions are no greater than with the usual construction, and are not objectionable.

centre at the middle position of the cross-head pin, strike arcs cutting the crank circle at c and d. Before cut-off in the forward stroke, the crank shaft, and with it the eccentric, must turn through the angle Ac, and in the return stroke Bd. Space off $a'e$ equal to Ac, and $b'f$ equal to Bd. Draw eg and fg, and we have point h, where the eccentric centre must be for cut-off at c, and i where it must be for cut-off at d. With radius equal to the length of the eccentric rod, and with centres at b, i, strike arcs meeting at k, and with same radius and centres a, h, strike arcs meeting at l. Now for admission at A and cut-off at c, the eccentric rod pin must be in the same position, and as the eccentric rod is of fixed length, this position must be l, that being the only point whose distance from both a and h equals the length of the eccentric rod. Similarly for admission at B and cut-off at d, the pin must be at k. The pin can be brought to these positions at the proper time by introducing a rock shaft in the valve motion having its centre at any point o; such that an arc struck from it shall pass through k and l, and then connecting the eccentric rod to it as shown. The valve stem should then be connected to the rocker, as shown at m, n. The eccentric rod positions for crank positions A, B are shown at al, bk. The rocker fulcrum might be located above the centre line, if preferred, at o'.

In cases where the valve chest is located on the top of the cylinder, a rocker of different type, with the arms on opposite sides of the fulcrum, becomes necessary. The construction for this type of rocker is essentially the same as shown in Fig. 32, which is lettered to correspond with Fig. 31. Of course, with this type of

rocker the eccentric positions *a, b* change places as shown. One result of the equalization growing out of the inequality of the rocker arms is to alter the port opening from that determined upon in the original designing of the valve. The motion as thus far determined should therefore be treated as a trial result only, and the dimensions of the valve and eccentric should be altered in the light of the experience gained.
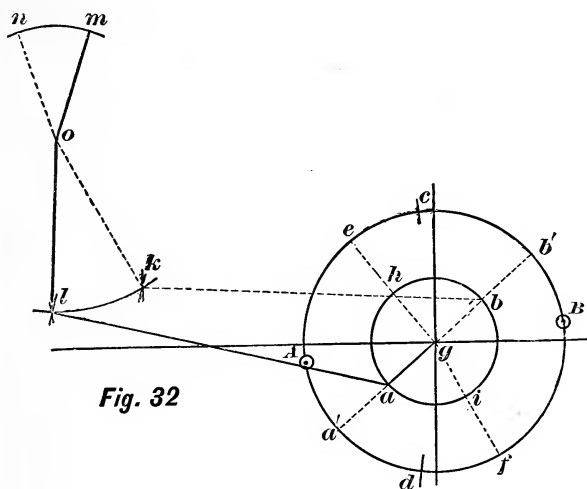
Fig. 32

At the close of the description of Fig. 31 it was stated that the rocker fulcrum might be located indifferently at either *o* or *o'* of that figure. This is strictly true so far as relates to equal lead and cut-off, but there is still a difference in the effect of the two positions. By suitably locating the fulcrum the compression and exhaust can be equalized for the two ends of the cylin-

der—exactly if the valve have no inside lap, and ap-
proximately if it have such lap. This has not the
unique interest which belongs to the equalization of lead
and cut-off, since it can be accomplished by other
means; but it forms an interesting study, nevertheless.
The method of accomplishing this equalization is
shown in Fig. 33, which follows the construction of Fig.

Fig. 33

31 up to and including the finding of k, l, but with lead
zero. Suppose, in the first instance, that the valve has
no inside lap, and by the methods already described
find the points of the piston stroke m, n, where release
and compression should occur, and by arcs whose com-
mon radius is equal to the length of the connecting rod
find the corresponding crank positions s, p. Lay off
Acs from a' giving q, and Bdp from b' giving r. Draw qg
and rg giving t and u, where the eccentric must be for
the two equalized compressions. With radius equal to
the eccentric rod, and centres t, u, strike arcs meeting in

v. Now locate the rock shaft fulcrum at *o*, such that the eccentric rod pin shall pass through *k*, *l*, and *v*, and the result will be a valve motion giving equal lead, cut-off, release, and compression. If the valve have inside lap, then, instead of one point, *v*, there will be two, just as with outside lap there are two points, *k*, *l*. In that case it will be found impossible to so locate *o* that the eccentric rod pin shall pass exactly through all four points. It should be then made to pass through *k* and *l*, and the difference be divided between the two points *v*. The release and compression will then be as nearly equalized as is possible.

SETTING THE SLIDE VALVE.

As the parts of an engine valve motion are assembled two dimensions are lacking: 1st, the angular location of the eccentric relative to the crank; and, 2d, the length of the valve rod. The eccentric is capable of being located in any angular position, and the length of the valve rod is usually capable of adjustment by means of jamb nuts each side of the valve, or some equivalent means. The setting of the valve involves locating the eccentric and fixing the valve at the proper point on the rod. There are two distinct steps to the process:

I. Locating the engine exactly on the centre;

II. Locating the eccentric and valve.

To locate the engine on the centre, proceed as follows: Turn the crank to any convenient distance above the centre, Fig. 34. Upon the side or face of the crank disc or fly-wheel, as most convenient, scribe an arc *a* by means of a tram *b* swinging from any conven-
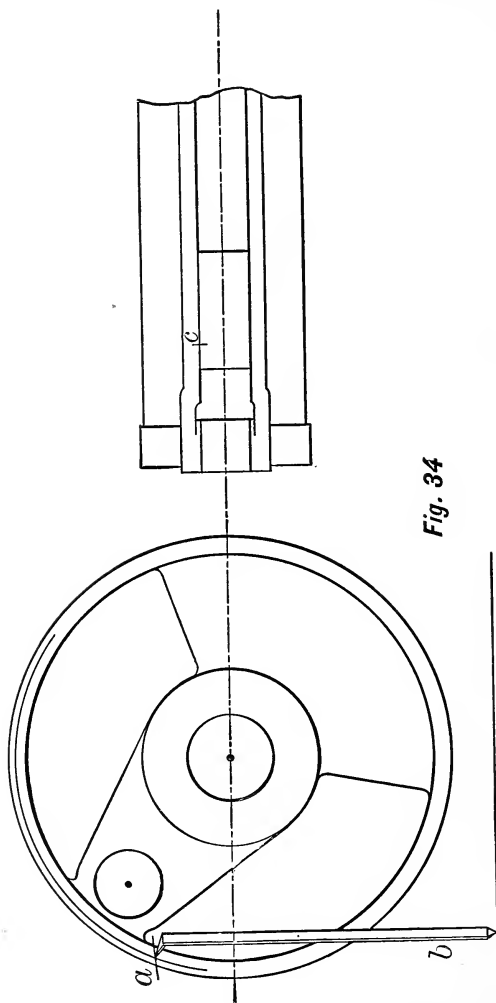
Fig. 34

ient fixed point on the engine frame or floor. Also scribe a line *c* on cross-head and guides. Turn the crank below the centre as shown by Fig. 35, the cross-head line receding from its mate on the guide and approaching it again. When these lines are exactly fair, stop the motion and scribe a second line *d* on the wheel, line *a*, now occupying the position shown in Fig. 35. With the dividers find point *e*, dividing the arc *ad* in halves. When point *e* is brought fair with the point of the tram, Fig. 36, it is clear that the engine will be on the centre. Repeat this construction for the other centre. One precaution is necessary in relation to the above, in order to obviate any error that might arise from looseness in the crank pin and cross-head pin bearings : In scribing the lines *a* and *d* have the crank pin pressing against the same brass for both lines. It matters not which brass be used, but the same one must be used for both lines. To locate the eccentric and valve proceed as follows : Locate the eccentric by the eye as near as may be, and ahead of its correct position rather than behind it. Bring the engine to either centre, as found above, *turning it in doing so in the direction of the proposed rotation* in order to neutralize any looseness in the connections. With the engine on the centre, locate the valve to give the required lead, after which turn the engine *in the direction of its future rotation* to the opposite centre. If the eccentric is ahead of its correct position, the lead for this position will be greater than the first ; if the eccentric is behind, the second lead will be less than the first, and probably negative. In either case the valve is to be adjusted on the rod to divide the differences in the lead. This being done, the valve is cor-
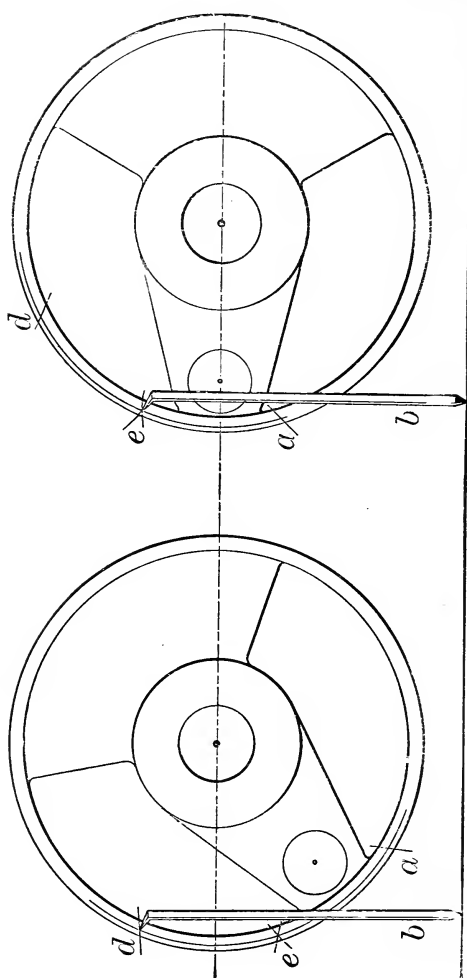
Fig. 36

Fig. 35

rectly located on the rod, the lead is equal at the two ends of the cylinder, but is too large or too small at both. To correct this it only remains to adjust the eccentric, *moving it in the direction of the rotation* until the valve have the proper lead. Verify the results, and the work is done.

Part II.

THE SLIDE VALVE WITH SHIFTING
AND SWINGING ECCENTRIC.

The Slide Valve with Shifting and Swinging Eccentric.

THE SLIDE VALVE AT SHORT CUT-OFF.

The difficulties which impede the use of the plain slide valve at short cut-off have been explained at length in Part I. Before explaining the shifting eccentric automatic valve gear, it is necessary to show how these difficulties have been surmounted. By referring to the section on the Limitations of the Plain Slide Valve those difficulties will be seen to be—

I. Premature release and compression—either of which, however, can be made later at the expense of making the other earlier still.

II. Inadequate port opening to steam or, in lieu of that, excessive size and travel of valve.

The first difficulty has been met by increasing the speed of the engine. All of the engines employing this description of valve gear are of the "high speed" type. In such engines a heavy cushion is appropriate and necessary to bring the reciprocating parts quietly to rest at the centres, and hence the early compression ceases to be a radical objection. Indeed, inside lap is

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given to the valve in order to delay the release—thereby, as has been explained, still further increasing the compression.

The second difficulty is met by two expedients, the first being sometimes employed alone, but more often in connection with the second. These expedients are, 1st, the use of balanced valves, usually of the true piston type or of the " pressure plate" type, both being perfectly balanced against the steam pressure ; 2d, the use of valves having multiple ports, by which the necessary throw of eccentric is halved or even quartered. The use of balanced valves permits the use of valves of large size and great throw; and the use of multiple ports gives sufficiently large openings with such throws as it is practicable to use.

It is believed that the first engine to embody the above features in connection with a shifting eccentric and a shaft governor was the Straight Line ; and hence that engine is entitled to be recognized as the progenitor of a large and vigorous family. So far as known, these features were first combined in an engine designed by Professor John E. Sweet, built at the Cornell University shops, and exhibited at the Centennial Exhibition.

That the difficulty of restricted port opening is a real one, may be gathered from any indicator diagram from a locomotive with plain valve at good speed and well "notched up." Such diagrams invariably show a marked fall in the steam pressure on entering the cylinder ; and it is largely on this account that such persistent attempts have been made to improve the locomotive valve motion. An appropriate introduction to

the study of multiple ported valves is found in one not necessarily balanced, which was designed more especially for use on locomotives,—to which it has been largely applied,—namely, the Allen valve, shown in Fig. 37. In·this valve the seat is shortened and a supplementary port *aa* is cast through the valve. This port registers with the end of the seat as shown in the figure, which represents the valve open by the amount of its lead. The course of the steam is shown by the arrows, from which it will be seen that the opening at *b* is added
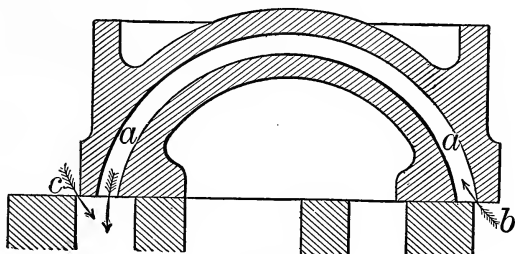

Fig. 37
The Allen Valve.

to the usual one at *c*; and that up to the point where the opening at *c* is equal to the width of passage *a*, the total opening is just twice what it would be with the usual form of valve.

The "pressure plate" type of valve is well shown in Fig. 38, which represents the valve of the Straight Line engine. The pressure plate *AA* receives the pressure of steam upon its back. It is prevented from pressing the valve proper to its seat by means of distance pieces above and below the valve, and slightly thicker than the valve. Recesses in the plate form in it an exact coun-
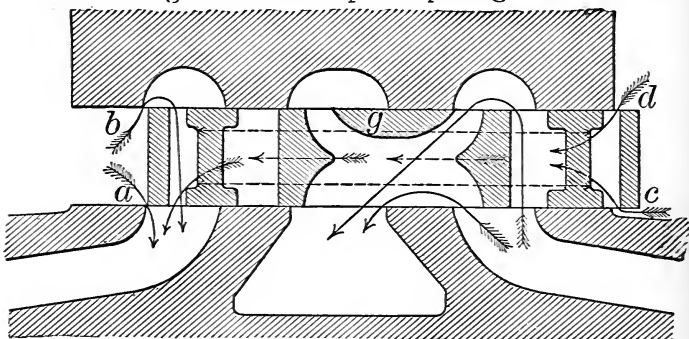
terpart to the valve seat. The valve slides between
the seat and plate like a square piston relieved of all
pressure. This valve, like those that follow, is shown
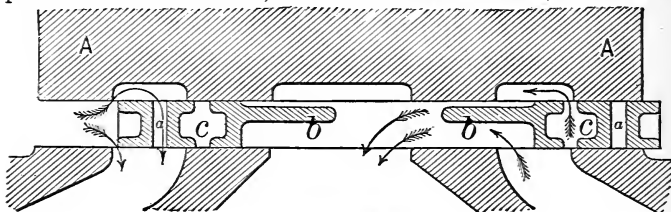
Fig. 38
The Straight Line Valve.

open to its lead ; and the manner in which the recesses
in the plate and the passages *aa* through the valve
combine to give a double port opening will be seen

Fig. 39
The Woodbury Valve.

from the arrows. The ledges *bb* are for the purpose of
protecting the finished surfaces of the pressure plate
from the cutting action of the exhaust steam. Some
designers of double ported valves have thought it best
to provide double ports for the exhaust, while others

have not. The illustrations of both the Allen and Straight Line valves will show that the opening to exhaust is amply large without double ports; but such ports increase the *quickness* of opening to exhaust, and so secure a desirable advantage. The valve under discussion is provided with them at *c, c*. Their action is
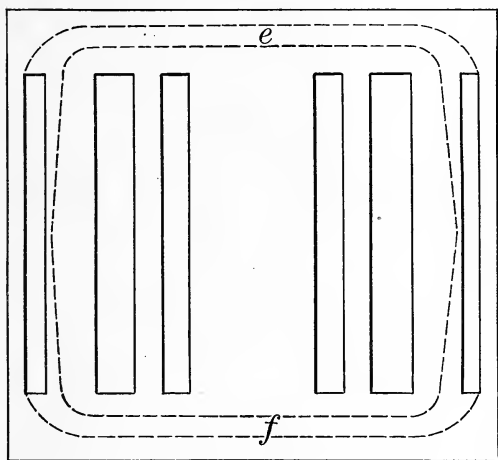
Fig. 40

precisely the same as that of the steam passages *a, a,* and need not be explained further.

Fig. 39 shows the valve of the Woodbury Engine Company. It combines the method of action of the Allen and Straight Line valves, and so secures four port openings to steam and two to exhaust. Openings *a, b* act precisely like those of the Straight Line valve, and openings *c, a* act substantially like the supplementary port of the Allen valve. This will be seen more clearly by re-

ferring to Fig. 40, which represents a plan of the Wood-
bury valve. Passages e, f are cast through the valve to
act in conjunction with the openings c, d of Fig. 39, in
the same manner that the passage aa of Fig. 37 oper-
ates in conjunction with opening b of the same figure.
Ledge g acts to protect the finished surfaces of the
cover plate from the action of the exhaust in the same
manner as ledges b, b of the Straight Line valve.

Fig. 41 illustrates the valve of the Armstrong engine,
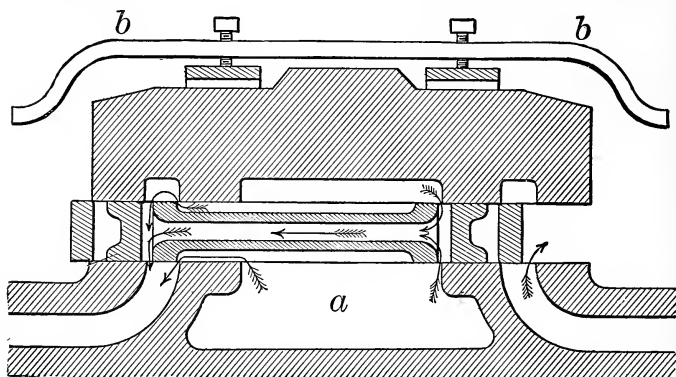
Fig. 41
The Armstrong Valve.

in which the action of the steam and exhaust edges of
the valve is reversed from the usual practice. The
steam enters at a, and the outer edges of the valve con-
trol the exhaust. The steam pressure tends to lift the
cover plate, and it is therefore held down to its seat by
means of the bridle b. The action of the steam ports
will be seen from the arrows. This valve gives four
openings to steam and two to exhaust.

The Rice valve (Fig. 42), like the last example, takes

steam from the inside. It gives two openings to steam and two to exhaust. The relief plate *aa* is in this instance a piston fitting the cylinder *bb*, this cylinder be-
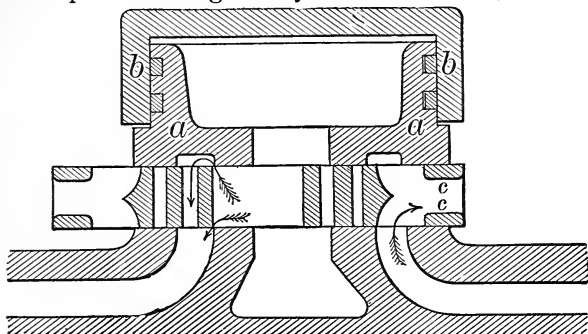
Fig. 42
The Rice Valve.

ing bolted to the floor of the steam chest. The pressure of steam within this cylinder forces the piston toward the valve, which, however, it is prevented from
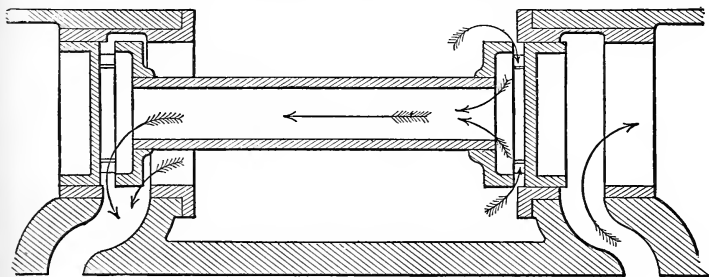
Fig. 43
The Armington and Sims Valve.

touching by means of distance pieces slightly thicker than the valve, and similar to those already described in connection with the Straight Line valve.

Fig. 43 illustrates the Armington and Sims valve, which is a true piston valve with double ports. The course of the steam is shown as heretofore by the arrows.

Fig. 44 shows the Ide double ported valve, which, like the last, is a piston valve.
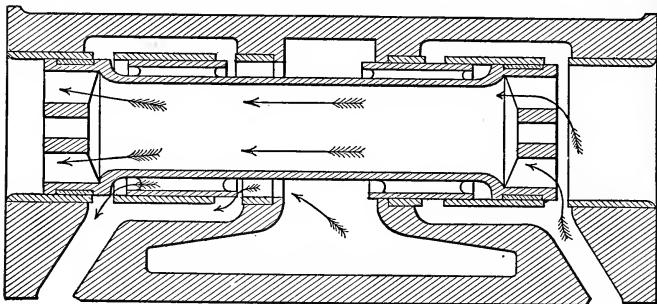

Fig. 44
The Ide Valve.

Of an entirely different type is the Giddings valve as used in the Russell & Company engine. This valve is shown in Fig. 45. Steam enters at a and exhausts at bb. Each end of the valve acts in much the same manner as the Allen valve, as will be understood from the arrows, while over all is cast a case c. The steam entering from the inside of the valve, its pressure would, if not counteracted, lift the valve from its seat. This is prevented by the use of "needle ports" (not shown), one connecting the live steam space within the valve to the body of the valve chest, and the second connecting the chest to the exhaust. The action of these ports is explained by Mr. Giddings as follows: "The steam is taken under the valve, which would result in throwing the valve off

the seat. This must be counteracted by pressure on the back of the valve sufficient to overcome the tendency to leave the seat. This I obtain by the small needle port communicating with the live steam passage on the inside of the valve. If there were no outlet this would soon result in an excess of pressure nearly equivalent to steam pipe pressure on the back of the valve, which would produce a hard working valve. To avoid this, I put another needle port opening, communicating with one of the exhaust 'D's' of the valve.
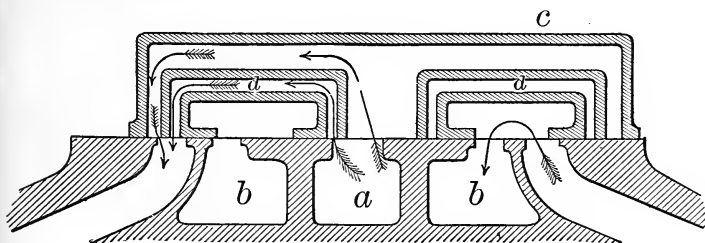
Fig. 45
The Giddings Valve.

The resulting pressure due to these two openings is just sufficient to overcome the tendency to leave the seat."

A remarkable feature of this valve, not attained by any other so far as known to the author, is the use made of the supplementary passage *d*. After the compression has commenced, and before opening to admit live steam from the steam supply, this passage opens into communication with the regular steam port. This increases the volume into which the steam is compressed, without, however, increasing the clearance space from which the steam is exhausted, since this supplementary

port is never in communication with the exhaust.   This is described by Mr. Giddings as follows :

" We find by increasing the capacity of the carry over port or portchamber, that we can use it as a reservoir or port into which to pack the surplus compression of a single valve automatic movement, thereby giving us a peculiar offset in the compression curve, and giving us from 10 to 12 per cent increase of area, and a consequent increase of power from a given sized engine. The amount of this is entirely within our control by a
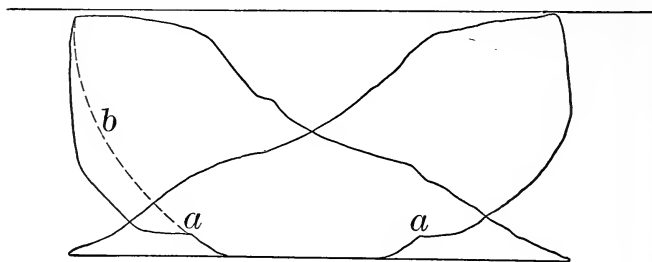
Fig. 46

variation of the cubical capacity of this passage way, thereby enabling us to get the compression curve down in the corner, something after the manner of the four valve engine cards."

This feature is of such unique interest that indicator-cards from one of these engines (Fig. 46) are introduced to illustrate it.   The offset mentioned will be seen at *aa*, the dotted line *b* representing the compression curve that would have resulted but for this provision.

The shifting eccentric valve gear, in connection with which all these valves are used, requires that the valve

shall have an excess of port opening and travel at the late cut-offs in order that they may be sufficient at the early cut-offs. Inspection of any of the valves illustrated will show that with large travels the supplementary port becomes closed after the main port is well opened ; and with such travels the effect of the supplementary port is merely to increase the quickness of port opening and closing.

* EQUAL LEAD AND CONSTANT LEAD.

The lead of a valve may vary in two ways, and to prevent ambiguity it is necessary to define and follow an exact use of terms. The reader is asked to note carefully the following explanations of the terms equal lead and constant lead. They will be used hereafter strictly as defined, and it is necessary that they be clearly understood.

Equal lead implies that the lead is alike at the two ends of the cylinder. Constant lead implies that the lead does not change for different grades of expansion. An engine might have equal lead for one grade of expansion and unequal lead for another grade, or the lead might be equal at all grades without being constant ; that is, the lead at the two ends of the cylinder might be always alike, but larger at both ends for three quarters cut-off than for one quarter. So, also, an engine might have constant lead without equal lead ; that is, the lead might not change for different grades of expansion, but at the same time be always larger for one end of the cylinder than for the other.

The above distinctions are radical and important,

and it is necessary that they be clearly seen in order to
understand what follows.

THE SHIFTING ECCENTRIC.†

The expedients which are employed for making what
is geometrically the plain slide valve available for early
cut-offs having been explained, it remains to describe
how the same valve can be made available for different
cut-offs.

In Fig. 47 let the circle *abc* represent the eccentric
path of a given eccentric in the Bilgram diagram, the
eccentric centre being at *d*, and *L* and *l* being, as usual,
the steam and exhaust lap circles. Let the centre of
the eccentric be shifted from *d* to *d'*, *dd'* being a
straight line perpendicular to *ac*. There will thus be
formed a new eccentric path *a'b'c'*. Laying off *b'd'* up-
ward from *c'*, will locate *Q'*, the new centre of the lap
circles.   From this centre the lap circles may be drawn
in the usual way, and from them the crank positions *A*
for the new point of cut-off, *B* for exhaust closure, and
C for release may be found. Since *bd* equals *cQ*, and
b'd' equals *c'Q'*, and since *dd'* is perpendicular to *ac*, it
follows that *QQ'* is parallel to *ac* : hence the lead open-
ing has not been changed. The lead angle, however,
has clearly been increased, and the port opening has
been reduced.   In the same way, the eccentric may be
shifted still further on the line *dd'*, and new points of
cut-off, release, and compression, and new values of the

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† Throughout this and the following section the angularity of the
connecting and eccentric rods is neglected.

port opening, found. Finally, if the eccentric be shifted to the position $d°$ on the line ac, the point Q will be located at $Q°$, when the port opening will be reduced to the lead opening and the cut-off will take place as much after the centre as the lead did before it. In all posi-

Fig. 47

tions the lead opening will be constant. If the valve have multiple ports, when the travel becomes so reduced as to bring them into action, the result will be to give double or quadruple the opening shown by the diagram, as the case may be. Should the movement of the eccentric be continued below the line ac, the result would

be to reverse the direction of the rotation. The study of reversing engines is, however, beyond the scope of the present work.

An eccentric arranged for adjustment on straight line *dd*°, as in this illustration, is called by the author a *shifting* eccentric.*

So far as known to the writer, there are but two engines in the American market which employ a shifting-eccentric. These are the Armington and Sims and the Russell (Giddings) engines. The former obtains practically a straight line motion of the eccentric centre by means of a combination of two eccentrics, while in the Russell engine the required motion is obtained by means of a straight guide keyed to the shaft and appropriate wings attached to the eccentric.

THE SWINGING ECCENTRIC.

Instead of shifting the eccentric across the shaft in a straight line as in the last examples, most designers have preferred to swing it by an arm cast in one with itself, and pivoted to an arm in the fly-wheel or other convenient piece. Such an eccentric is called by the author a *swinging* eccentric, and its effect upon the steam distribution, as distinguished from the shifting eccentric, is to vary the lead opening at different points of cut-off. The nature and degree of this variability depends in large degree upon the location of the pin from which

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* For want of another word, the term *shifting eccentric* is also used as a general expression which includes both shifting and swinging eccentrics. This double use of the word will not, however, cause confusion.

the eccentric is swung. If the pin is on the same side
of the shaft as the crank, and in the centre line of the
crank, as in Fig. 48, the path of the eccentric across the
shaft is the arc $dd°$. The path of the point Q will then
be a similar arc $QQ°$, with the same radius and with its
centre located in the line ef. It is clear from the posi-
tions of the various lap circles that the lead opening
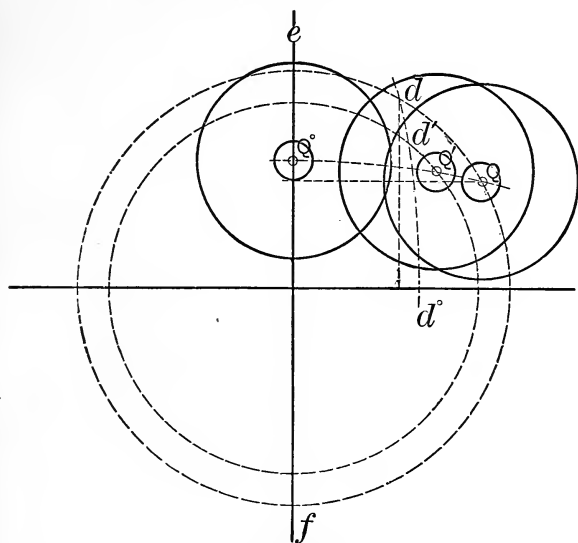
Fig. 48

will not be constant with this arrangement, but will be
greater in the early cut-offs than in the late ones. The
action upon the other events of the stroke is substan-
tially the same as that of the shifting eccentric. With
the pin located in the centre line on the opposite side
of the shaft from the crank, as in Fig. 49, the arc in
which the eccentric swings has its convexity reversed

from the last figure. The path of the point Q will also be reversed, and instead of receding from the centreline in the early cut-offs, it will approach it, thus diminishing the lead opening in those cut-offs. This diminution of lead may be carried so far as to make the lead zero or even negative in the early cut-offs—a fact which
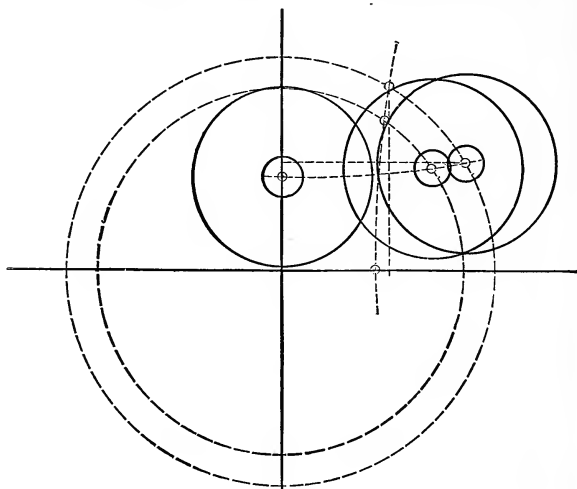
Fig. 49

will be shown on the diagram by the earlier steam lap circles crossing the horizontal centre line. The same result of a decreasing lead in the early cut-offs can be obtained with the eccentric swing pin on the same side of the shaft as the crank, but raised above the centre line as in Fig. 50, in which the swing pin is located on the line dg extended. Similarly, by raising this pin when located opposite to the crank, an increasing lead can be obtained. By locating the pin half way between
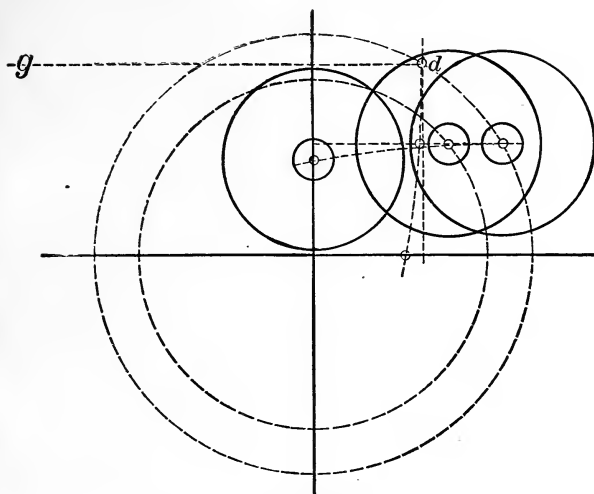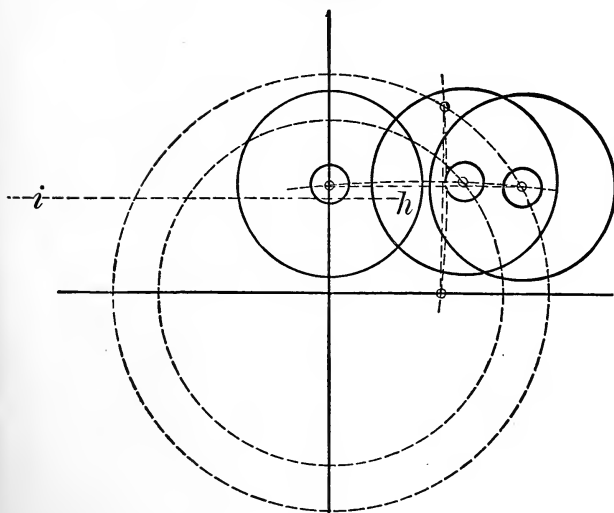
Fig. 50

Fig. 51

the positions of Figs. 48 and 50, on the line *hi* of Fig. 51 the lead will be the same at the smallest as at the greatest throw ; and by suitably placing it as in Fig. 52, the lead can be made alike for any two expansions desired. In this construction the two points of cut-off
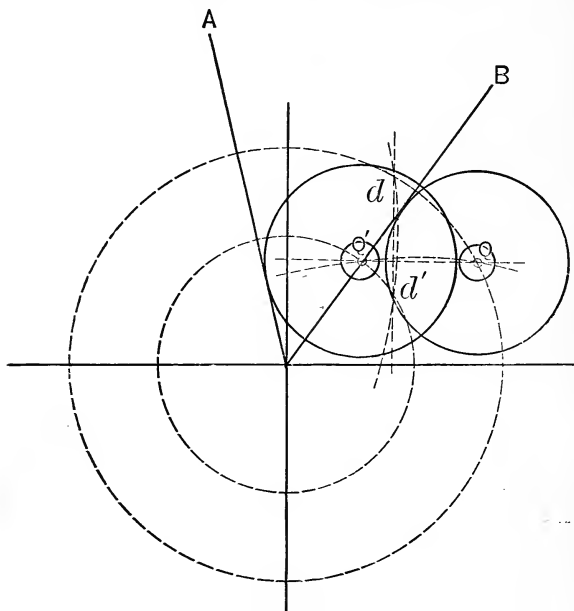
Fig. 52

for which equal lead is desired are decided upon, and the corresponding crank positions A, B are drawn. The lap circles are drawn in the usual way for A and B, the lead being made the same for both. Points Q and Q' are then transferred to d and d', and the eccentric is so hung that its centre shall pass through the points d and d'.

As has been pointed out, if the valve have a negative lead in the early cut-offs, it will be shown in the diagram by the lap circle going below the horizontal centre-line. In this case if the cut-off be made early enough the lap circle will pass through the centre of the shaft. This marks the point where the port opening becomes zero by reason of the eccentric throw becoming reduced to equality with the lap. For any smaller throw there is no admission whatever.

Designers have usually endeavored to obtain as nearly a constant lead as possible. The author considers, however, that for stationary engines, where the speed is fixed, a lead decreasing in the early cut-offs is more suitable. That the lead should be equal at the two ends of the cylinder, there is, however, no question. This entire subject will be discussed at greater length in the section on Equalized Lead.

* THE ANGULARITY OF THE ECCENTRIC ROD.

Thus far in Part II the influence of the angular vibration of both connecting and eccentric rods has been ignored. It will be understood that primarily both connecting and eccentric rods produce the same distortions with the shifting as with the fixed eccentric. There is, however, with the shifting or swinging eccentric an additional distortion produced by the angularity of the eccentric rod growing out of the fact that the vibration of that rod varies in amount with the varying throw of the eccentric—the small throw due to an early cut-off giving a small angular vibration, and the large throw due to a late cut-off giving a larger vibration.

Taking up first the case of the shifting eccentric in Fig. 53,* let *A, B* represent the crank pin of a shifting eccentric engine when on the dead centres, *a, b* being the corresponding positions of the eccentric centre at its greatest throw, *h, i* at its mean throw, and *e, f* at its smallest throw. The positions of the eccentric rod for mean throw of the eccentric are shown at *ch, di,* and the valve in both positions is open by the amount of its lead, as shown by the upper valve sketch for *c,* and the lower one for *d.* Let the path of the eccentric
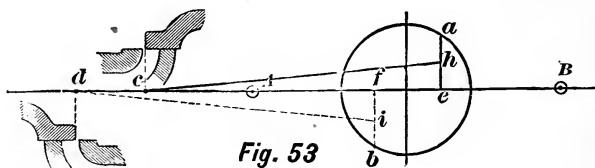

Fig. 53

centre, when shifted across the shaft to change the expansion, be a straight line occupying the position *ae* for crank position *A,* and *bf* for crank position *B.* Imagine the crank on the centre *A,* and shift the eccentric toward *e.* Obviously point *c* will be moved to the left, and the lead will be disturbed. For crank position *B* the same is true; and what is still worse, while the movement for crank position *A* decreases the lead, that for *B* increases it. If, with crank at *A,* the eccentric be shifted towards *a,* and with crank at *B* towards

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* See foot-note, page 55.

This feature of the present diagrams shows a greater irregularity in the lead in the usual form of construction, as well as in the form to be described, than actually obtains with working proportions. This, however, for the present purpose, is rather an advantage than otherwise.

b, the lead at the two points will be disturbed in the op-
posite directions; i.e., for position *A* the lead will be in-
creased, and for *B* decreased. The broad fact is evi-
dent, that, owing to the varying angularity of the ec-
centric rod, an engine laid out as shown in Fig. 53 could
not have a constant lead, and it could only have an
equal lead for some one (selected) grade of expansion.

With a swinging eccentric, the simplest case is where
the eccentric swings about a point which, with crank
at *A*, Fig. 54, coincides with *c*. Thus, suppose a
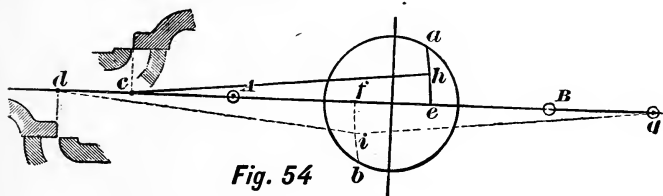
Fig. 54

large disc keyed to the shaft, and arrange the ec-
centric to swing about a pin fixed to the disc, the
centre of the pin for crank position *A* coinciding
with *c*, Fig. 54. Now shift the eccentric from *h* to-
wards *e* or *a*, and *c* will not be disturbed. When,
however, the crank is at *B*, the pin *c* will be at *g*; and
if the eccentric be shifted from *i* towards *f* or *b*, point *d*
will obviously be disturbed more than in the corre-
sponding movement of Fig. 53; and it may be said in
general, that with the eccentric rod arranged in the
common way, as shown in Fig. 53, any change in the
path of the eccentric across the shaft, to correct the in-
constant lead at one dead centre, will only make mat-
ters worse at the other, and by no possible modifica-

tion can the lead be made equal for more than one
grade of expansion.

The arrangement of Fig. 54, however, while of inter-
est and value in a theoretical study of the subject, is of
no practical importance, because its use would require
so large a disc for the attachment of the eccentric
swing pin as to be impracticable. It therefore be-
comes necessary to examine the situation with the ec-
centric swung from a position nearer the shaft. Let
the pin be located at the centre of the crank pin—a
common position—as in Fig. 55, the eccentric rod
being much longer than the crank, as it always is in
practice. In that case the actual path of the eccentric
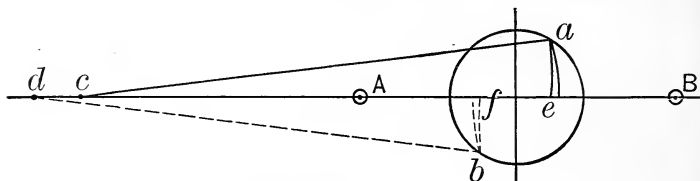
Fig. 55

for crank at *A* will be to the right of *ae* of Fig. 54, and
to the left of *bf*; consequently the lead will increase
at both ends of the cylinder for short cut-offs, but the
increase will plainly be greater for crank position *B*
than for *A*. The result is a lead increasing in the early
cut-offs, but increasing much faster for one end of the
cylinder than for the other, and hence equal at the two
ends of the cylinder for one grade of expansion only.
Similarly it might be shown that by swinging the ec-
centric from a point diametrically opposite the crank
pin the lead would decrease in the early cut-offs (see
Fig. 49), but decrease much more rapidly for one end

of the cylinder than for the other; and it is clear that whatever quality is sought for in the lead, and determined so far as the Bilgram diagram can do it, it will in fact be modified by the angular vibration of the eccentric rod. It seems unnecessary, however, to examine the subject in detail further.

* EQUALIZED LEAD.

So far as known to the author, the only engine in which any attempt is made to correct the distortions which have just been explained is the Straight Line. The valve motion of this engine, like its mechanical details, is an exhibition of refined ingenuity which it would be difficult to surpass. In the following it will first be explained how the engine was originally built to secure a substantially constant lead, and after that the present construction will be shown. The construction to be described is essentially the same as that already used in equalizing the cut-off of fixed-eccentric engines (page 54), and it should be understood that the present use of that construction was the original one—its use for equalizing the cut-off being in fact an offshoot of its original use for equalizing the lead. In equalizing the lead as well as the cut-off, two types of rocker are possible. The first type of rocker is shown in Fig. 56, the parts being lettered as in the three preceding figures. From h and i, hc and id are drawn parallel to the centre line, and each is made equal to the length of the eccentric rod. The rocker fulcrum is then so located at o that the pin for the eccentric-rod shall describe an arc passing

through c and d. The pin for the valve rod is located as usual, m belonging with c, and n with d. The eccentric is shifted by being swung from a pin, whose location for crank position A coincides with c. As explained in connection with Fig. 54, the disturbance of the lead for crank position A is thus eliminated. When the crank is at B, c is at g in line with id. In this position of the parts, shifting the eccentric on the line
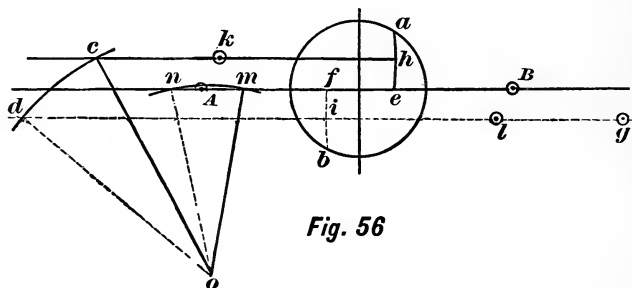
Fig. 56

bf will disturb the lead a trifle, though much less than in the constructions of Fig. 53 or 54. We have here, then, a construction which eliminates the disturbance for crank position A, and practically eliminates it for B, and thus secures substantially a constant lead. Comparing Figs. 54 and 56, the essential difference in the two plans is apparent. In Fig. 56, ch and di are parallel, and hence both c and g are in line with the eccentric rod position to which each belongs; whereas in Fig. 54 ch and di are not parallel, and hence, while c is in line, g is not, and cannot be made so. The construction so far explained, would, however, lead to inconvenient dimensions of some of the parts. The centre of motion for shifting the eccentric is therefore in practice moved

inward from c to some convenient point k on the line
ch, the eccentric rod pin still remaining at c. The po-
sition of k for crank at B is of course l. This change
introduces a trifling error for crank position A, and by
an equal amount increases the existing error for B; but
the final irregularity is infinitesimal as compared with
Fig. 53 or 54, and the mechanism accomplishes its ob-
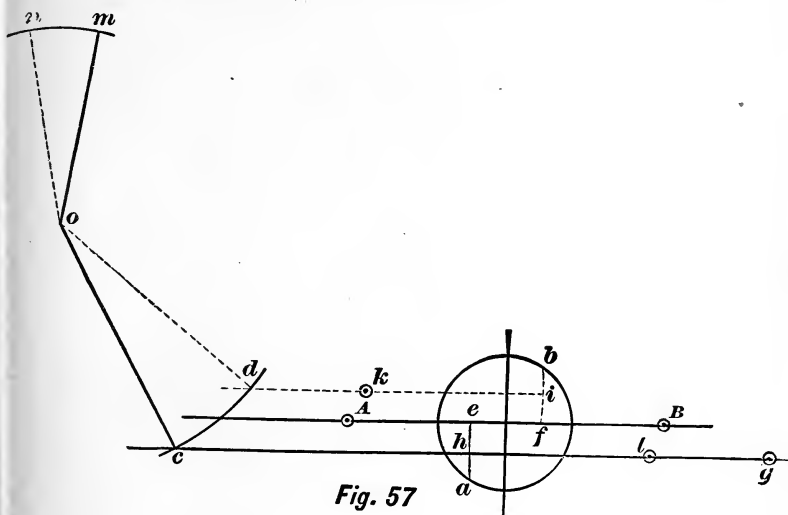
Fig. 57

ject—obtaining a practically constant and equal lead
at all grades of expansion.

The second type of rocker introduces no material
change in the lay-out. For an engine with the steam
chest on top of the cylinder it is illustrated in Fig. 57,
the parts being lettered to correspond with Fig. 56.
This type of rocker reverses the motion of the eccen-
tric, and hence positions a, b, change places as shown.

As the Straight Line engine is actually built, how-
ever, the steam chest is on the side of the cylinder, and
hence the second type of rocker takes the unusual
form of Fig. 58. There is, however, no change in the
essential principle of the construction, which is, that the
two positions of the eccentric rods which belong with
crank positions *A*, *B* shall be parallel to one another.
The object of using this form of rocker is as follows:

Fig. 58

This valve motion, like the usual form, when set for
equal lead, gives a larger port opening for one end of
the cylinder than for the other. It is well known that
the speed of the piston is faster in the end of the cylin-
der farthest from the crank shaft. Now the second
type of rocker gives the large port opening to that end
of the cylinder in which the piston travels the faster,

while the first form gives the reverse relation. Hence the choice in the construction.

As now built, however, the Straight Line engine has a decreasing lead in the early cut-offs, becoming in fact negative in the earliest grades. The reason for this change in practice is as follows : In shifting eccentric engines, as is well known, the compression increases as the cut-off grows shorter. The total cushion by which the momentum of the reciprocating parts is arrested is the sum of the exhaust cushion and the lead cushion. Now if the total cushion is to be constant at all points of cut-off, as it should be, the lead must decrease as the compression increases, and at the early cut-offs the lead should be negative. Furthermore, in such engines, as has been explained, both the release and compression for early cut-offs occur earlier than is desirable. Now by laying out two valves for the same cut-off, one with positive and the other with negative lead, it will be found that the valve with negative lead gives considerably later release and compression than does the one with positive lead. In other words, the introduction of negative lead at the early cut-offs, in addition to off-setting in a measure the increasing compression, prolongs the expansion, thereby getting more work out of the steam, and also delays the compression, thereby still further reducing the cushion. The result is accomplished as shown in Fig. 59, which is a modification of Fig. 58. The point k, instead of being on the line di, is placed above it. This brings l equally below ch, extended ; and, as will be seen by referring to the upper valve sketch for m and the lower one for n, reduces the lead for both crank positions A and B, as the eccen-

tric is shifted inward towards e and f. If the elevation
of k above di is sufficient, the lead at the early cut-offs
will obviously be negative. To fully appreciate the
merit of this construction, Fig. 59 should be compared
with Fig. 53, when it will be seen that while in Fig. 53
shifting the eccentric inward from h, i, decreases the
lead in the upper valve sketch, **it** increases it in the
lower one; in Fig. 59, on the other hand, shifting the
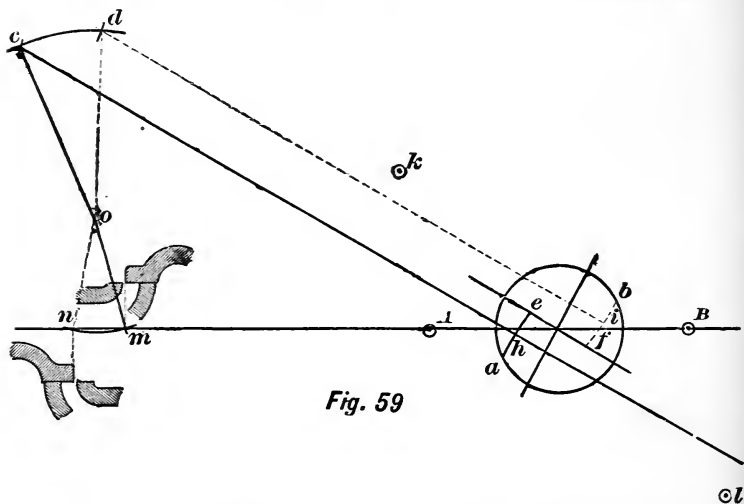
Fig. 59

eccentric inward from h, i, decreases the lead in both
valve sketches.

Again, Fig. 59 should be compared with Fig. 55, and
with the suggested companion to it having a decreasing
lead in the early cut-offs, when it will be seen that
while the plans are essentially alike in having an incon-
stant lead, they are unlike in that in Fig. 59 the lead
is substantially equal at both ends of the cylinder

throughout the range, but in Fig. 55 it is equal at one grade of expansion only.

* EQUALIZED LEAD AND CUT-OFF. †

It has been shown how, by different methods of proportioning the parts, the same mechanism can be laid out at will to give either exact equalization of the cut-off in fixed eccentric engines, or approximate equalization of the lead in shifting eccentric engines. It remains to be shown how a proportion of parts can be found which will satisfy both constructions, and thus obtain a practically constant and equal lead, an exactly equal cut-off for any chosen grade of expansion, and approximately equal cut-offs for all grades.

Referring again to Fig. 56, it appears that the fundamental principle of its construction is the diminution of the inclination to one another of the lead positions of the eccentric rod; and, referring to Fig. 31, it appears that the fundamental principle of its construction is the direct reverse of this, i.e., the increase of this inclination. It hence appears at once that the constructions of Figs. 56 and 31 are incompatible, and cannot be reconciled with one another. Comparing Figs. 57 and 32, on the contrary, it appears that in this general way they agree; but while in Fig. 57 the inclination of hc and id is reduced to actual zero, i.e., the rods are made parallel, in Fig. 32 al and bk are still inclined at an appreciable angle. Now the inclination of al and bk to one another in Fig. 32 can be varied at will by

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† First published by the author in the *American Machinist* for March 14, 1889.

changing the length of the eccentric rod; and by
choosing a proper length they can be made parallel,
and the proportions so found will satisfy the construc-
tion for equal lead, and for the particular grade of ex-
pansion for which it should be drawn for equal cut-off
likewise. For other grades it will, of course, satisfy the
construction for equal cut-off only approximately. One
qualification must, however, be made. In Fig. 57, A
and B are located at the dead points, while in Fig. 32
they are located at the points where admission occurs,
and these are not usually the dead points. It will
simplify matters to make these points coincide by con-
sidering, in the first instance, an engine with a lead of
zero. In Fig. 60 the construction of Fig. 32 is repeated
for the mean throw of the eccentric and with lead zero,
but with several lengths of eccentric-rod, giving a cor-
responding number of points k', k'', k''', l', l'', l'''. It is
plain that the inclination to one another of $k'b$ and
$l'a$ is less than that of $k''b$ and $l''a$, which in turn is
less than $k'''b$ and $l'''a$, the degree of inclination de-
pending on the length of rod used. If a length of rod
can be found such that its two positions shall be paral-
lel to one another, the rod so found will obviously sat-
isfy the constructions for both equal lead and equal
cut-off. This length is found in the following manner:
It is obvious that all the points k', k'', etc., are on the
straight line pg, and similarly l', l'', etc., are on qg;
therefore, draw pg and qg. Assume a trial length of
eccentric-rod, and with centres a, b, strike arcs giving
k, l, such that bk and al are parallel, repeating the
construction with different lengths of rod until the
correct length is found. Locate the rocker fulcrum so

that the eccentric rod pin shall pass through k and l.
Now positions al, bk obviously satisfy the construction for equal cut-off, and, being parallel, they also satisfy that for equal lead ; and an engine with its valvemotion laid out in this manner will have approximately
equal and constant lead, equal cut-off for that eccentric throw for which the construction is made, and an

Fig. 60

approximately equal cut-off for other positions of the
eccentric.

The construction of Figs. 56, 57, and 58 was made
with the crank on the centre, while that for Figs. 31
and 32 was made with the crank in position for admission of steam, and in Fig. 60 these methods were
reconciled by supposing the admission to occur on the
centre. No material error would be introduced if this

method were followed in all cases; but if it is desired
to follow the strictly correct method, it can be done in
the following manner : Fig. 61 is a reproduction of Fig.
32, constructed for the mean throw of the eccentric,
with some additions. Find points b_1, a_1 and k_1, l_1 cor-
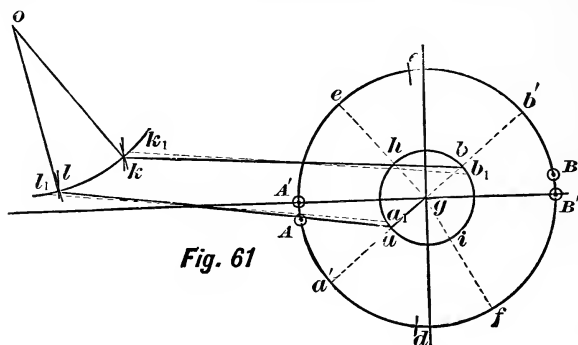
Fig. 61

responding with crank positions A', B', and draw $k_1 b_1$
and $l_1 a_1$. Now the construction of Fig. 32 gives the
positions of kb and la, while it is desired to make $k_1 b_1$
and $l_1 a_1$ parallel by suitably selecting the eccentric rod
length. This should be done by trial, as before, re-
peating the trial until the desired result is reached.

PART III.

THE SLIDE VALVE WITH INDEPENDENT CUT-OFF.
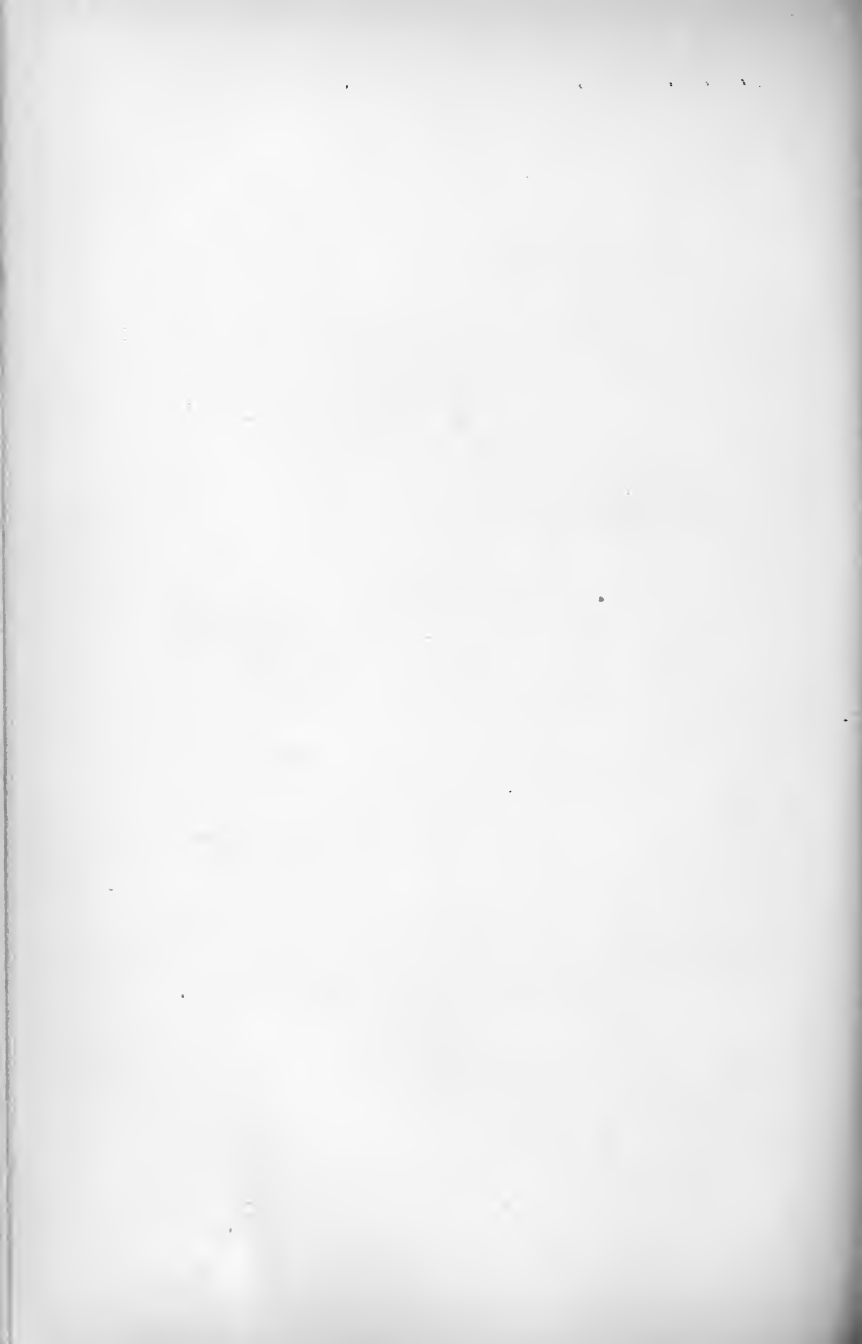
The Slide Valve with Independent Cut-off.

INTRODUCTORY REMARKS.

An early cut-off being a necessity for an economical use of steam, it comes about that with valves of the construction previously described the leading considerations in their design are those pertaining to the steam side of the valve. The valve and eccentric being designed with reference to the steam side, so as to secure an early cut-off, there is little that can be done with the exhaust side beyond reconciling conflicting requirements as well as possible. In engines provided with independent cut-off valves this condition no longer holds: the exhaust can usually be arranged to suit the designer's fancy; and it hence follows that in such engines the leading considerations in the design of the main valve are usually those pertaining to the exhaust. It is essential that the exhaust have a certain lead, in order that the cylinder may free itself of steam, and, on the other hand, this lead should be no greater than is necessary for this purpose, since that would involve exhausting the steam at a point where it might still do

some useful work. The determination of the point of release is, therefore, a leading factor in the design of this class of valves. It is not to be expected that there will be any close agreement in a detail of this character in the work of different designers; but, as a general rule, modified somewhat by questions of piston-speed, etc., it may be said that release should occur at from 93 to 95 per cent of the piston stroke.* The other event of the exhaust side of the valve, the compression, is determined, it must be owned, largely by the taste and fancy of the designer. A late compression by requiring a small exhaust lap conduces to a small travel of the valve, which, if it is to be unbalanced, is a desirable feature. The features of the exhaust side of the main valve and the port opening to steam having been settled, the steam side is determined by the force of circumstances. To give the proper points of exhaust opening and closure, the steam lap will usually be small and the cut-off late. This, however, is of no consequence, as the cut-off valve is introduced for the express purpose of providing for it.
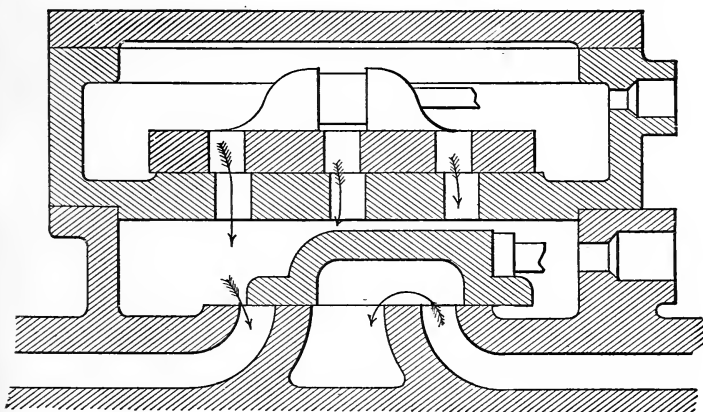
THE GONZENBACH VALVE GEAR.

A description and analysis of this valve gear is here given as an introduction to those which follow. It is now seldom employed. It comprises two valve chests, two valve seats, and two valves, as shown in Fig. 62. The lower or "main valve" is driven by a fixed eccen-

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* In engines of slow rotative speed—for example, Corliss engines and others of similar general character—the release will frequently be found to be somewhat later than the figure given in the text.

tric, and determines the admission, release, and compression of steam. The upper or "cut-off valve" is used solely for the purpose of the cut-off, and is usually of the gridiron type, in order to secure quickness of cut-off with moderate travel. It is driven by an eccentric of its own, which, if the expansion is to be varied, must be turned forward or backward, as the case

Fig. 62
The Gonzenbach Valve,

may be, on the main shaft. This movement affects the angular advance only, and, unlike the eccentric movements of Part II, does not change the travel of the valve.*

The action of this cut-off valve is different from anything that has thus far been examined. The previous

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* The following applications of the Bilgram diagram to the Gonzenbach, and also to the Buckeye and Bilgram valve gears, are substantially the same as those previously published (now largely inaccessible) by Mr. Bilgram.

valves open and close their ports with the same edge, i.e., in opening the port the valve draws to one side, and in closing it resumes the previous position. This cut-off valve, however, opens and closes the port by the port in the valve passing bodily across the port in the seat, the opening being done by one edge, and the closing by the other. Further, the same ports serve for both ends of the cylinder; the valve ports passing over the seat ports in one direction for one end of the cylinder, and in the opposite direction for the other end.

It will be seen from Fig. 62 that the cut-off valve

Fig. 63

Fig. 64
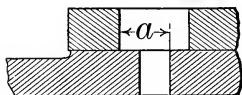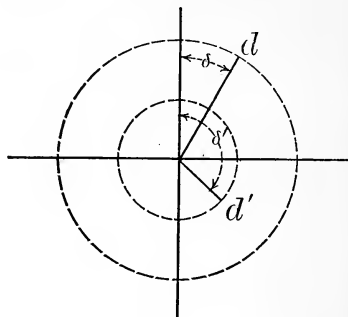
has negative lap, since the ports are open when the valve stands at its central position. The width of port in the valve may equal or exceed the width of port in the seat. In the former case, the negative lap is equal to the width of port; in the latter, to the distance a of Fig. 63. The location of the eccentrics is shown in Fig. 64, d being the main and d' the cut-off; δ being the advance angle of the former, and δ' of the latter. The centres of the lap circles Q and Q', Fig. 65, are found in

the usual way, by laying off the angles δ and δ' upward from the horizontal centre line. The effect of the three ported valve being equivalent to a valve with a single port of three times the width and travel, the throw of the cut-off eccentric is in the diagram in-
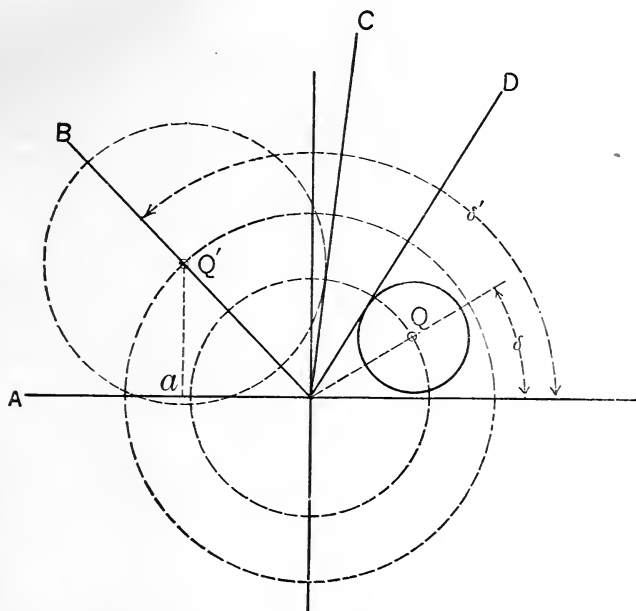
Fig. 65

creased to three times that shown in Fig. 64, and the lap circle is likewise drawn, with a radius three times the actual negative lap for each port. Starting at crank position *A*, it is clear that the main valve is open by the amount of its lead. The cut-off valve is at a distance $Q'a$ from its central position, which being

less than the negative lap, the cut-off ports are open,—
a fact which is also shown by the dotted (negative) lap
circle going below the horizontal centre line. As the
crank rises, the port opens more widely, up to position
B. In the original discussion of the Bilgram diagram
it appeared that when the crank passed through *Q* the
valve stood central upon its seat. In that position the
ports of the present valve stand wide open,—as is
apparent from the plan of the valve and the present
diagram alike. Passing crank position *B*, the valve
begins to close the port, not by returning toward its
former position as with previous valves, but by passing
on to the other side of its centre line,—as is indicated
in the present diagram, by the perpendicular from *Q'*
falling upon the opposite side of the crank.

The closure is completed and cut-off takes place at
C, which is indicated in the usual manner. From this
point expansion goes on in the *cylinder and lower chest
together*, until crank position *D* is reached, where the
main valve closes its port. As has been stated, the ex-
pansion is varied by changing the angular advance of
the cut-off eccentric. If δ' of Fig. 64 be increased, *Q'*
of Fig. 65 will be lowered toward the centre line and
the cut-off position *C* of the crank will be shifted to an
earlier part of the stroke. This change in the cut-off
will be accompanied by an earlier and earlier admission
from the upper to the lower steam chest, as will be
shown by the large lap circle having a larger and larger
segment below the horizontal centre line. Crank posi-
tion *D* extended backward gives the position at which
the main valve cuts off the steam on the previous stroke,
and it is clear that the cut-off eccentric might be ad-

vanced so far that the admission from the upper to the lower chest would occur before the main valve had closed its port in the previous stroke; i.e., in advancing the eccentric to obtain an early cut-off the result would be to give a second admission of steam at the latter end of the expansion. This feature limits the range of variation to the expansion which this gear can give. The only way to provide a shorter cut-off is to increase the lap of the main valve, since this carries crank position D backward, and allows the cut-off lap circle to be carried farther back before interfering with the main valve.

The principles of this valve, and of the application of the diagram to it, can be fixed in the mind by following the solution of

Problem VII. A Gonzenbach valve gear is to be constructed with a maximum cut-off of $\frac{5}{8}$ stroke. The greatest port opening to steam of the main valve is to be $1\frac{1}{4}''$. Since there is nothing to prevent liberality in this respect, the cut-off ports will number three, each one inch wide in valve and seat alike. Required the shortest possible cut-off, and the positions of the cut-off eccentric for the earliest and latest cut-off.

In Fig. 66 the centre Q of the main valve lap circle is found in the usual manner. Lead position A and maximum cut-off position B are then drawn. At the latest cut-off the cut-off valve lap circle must be tangent to A and B, and its radius being three inches (three times the port opening), it is easily drawn, giving Q' the position of cut-off eccentric for latest cut-off. Extending B downward and finding Q'' such as to make the cut-off lap circle tangent to B extended,

determines crank position *C,* the earliest cut-off prac-
ticable with the dimensions given, and also the range
of movement *Q'* to *Q''* of the cut-off eccentric.

It has been shown that the range of cut-off with this
gear is somewhat limited. Another defect of the ar-
rangement is the large volume of the lower chest, which
increases the clearance space during expansion. The
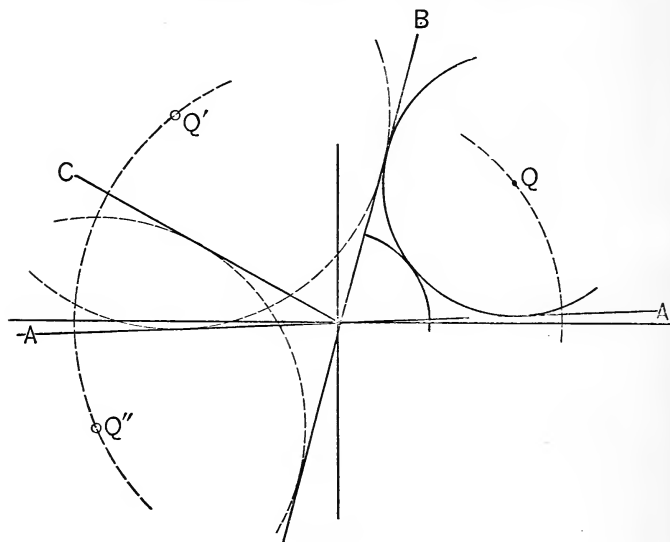
Fig. 66

object of a separate cut-off valve is to introduce early
cut-off, and it is in these early cut-offs that the effect
of this increased clearance is greatest, making a serious
discrepancy between the " real " and " apparent " ex-
pansion. For these reasons, together with the inac-
cessibility of the lower valve, the plan has largely fallen
into disuse.

THE MEYER VALVE GEAR.

In this gear, which has been very extensively used, a separate valve is used for the sole purpose of cutting off, as in the last example. The general arrangement of the valves is shown in Fig. 67, from which it will be seen by reference to the dotted lines that the main valve is essentially a plain valve of the usual type, with the addition of a bridge at each end to form a port through
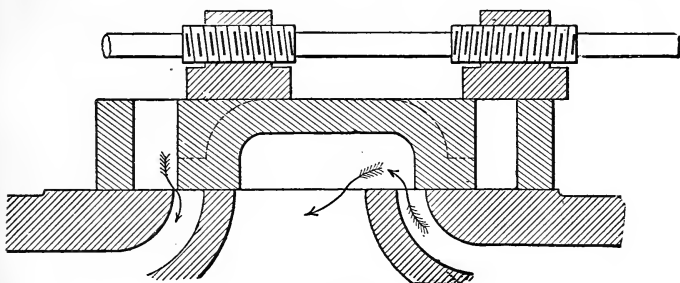
Fig. 67
The Meyer Valve.

it, and planed upon its back to form a seat for the cut-off valves. These cut-off valves are driven by a separate eccentric, and, as shown, the cut-off valve rod contains a right and a left hand screw upon which the valves are threaded. The valve rod has a hand wheel upon it, and its connection with the eccentric rod is such as to permit its being rotated at will by means of the hand wheel. This rotation increases or decreases the distance apart of the valves, and thereby changes their lap and the point of cutting off. An index is attached, which, moving over a graduated scale, shows at a glance the position of the valves upon the stem and

the degree of expansion. Occasionally this rotation of the valve rod has been connected to the governor, but the extent of the movement required and the friction incident to the mechanism are so great as to render such a plan a questionable success. Unlike the previous gear, the angular location of the cut-off eccentric is not a matter requiring exactness. In the older practice it was customary to place that eccentric exactly opposite the crank, or, since that gave the same motion, to connect the valve rod to the cross-head by means of a lever. This plan is still followed in marine, hoisting, and other engines which are to turn in both directions, since the motion of the cut-off valves is then correct for both forward and backward rotation. In present practice the cut-off eccentric of stationary engines is not usually placed so far in advance of the main eccentric. A common location is forty five degrees in advance. The effect of this is to require a smaller movement for a given change of the expansion. To offset this advantage, it reduces somewhat the width of port opening given by the cut-off valve and the speed of cutting off.

The application of the Bilgram diagram to this gear is shown in Fig. 68. The locations of the eccentrics are shown at *d* and *d'*, the throw of the latter exceeding that of the former, as is customary in practice. It will be understood at the outset that since the lap of the cut-off valve is changed to vary the point of cutting off, a number of cut-off lap circles will appear in the diagram. Some of these will represent positive and some negative lap, and the determination of the proper lap for different expansions is one of the leading points to be determined from the diagram. Since the seat of

the cut-off valves is upon the back of the main valve, it is clear that the diagram must show the position of the former *in relation to the latter.* Beginning with crank

Fig. 68

position *A*, it is clear that the main valve is advanced to the right of its mid position by the distance Qa, and the cut-off valve by the distance $Q'a'$. The cut-off valve

is therefore removed from its mid position a distance
$Q'a''$ greater than the main valve.   If it is desired that
the cut-off shall take place with the crank at A, there
must be a lap given to the cut-off valve equal to $Q'a''$.
Hence the lap circle l'.   Similarly for crank position B
the main valve is at a distance Qb and the cut-off $Q'b'$
from the central position.   The displacement of the
main valve is here the greatest by $Q'b''$, and if the cut-off
valve had a lap of zero the main valve port would still be
covered by the distance $Q'b''$.   Therefore, if the cut-off
is to occur at this position of the crank, the cut-off valve
must have a negative lap equal to Q', b''.   Similarly the
value of the lap required for any cut-off may be found.
If the range of cut-off is to be from zero to that given
by the main valve, the positive lap required for the
former and the negative lap for the latter is easily found,
and the sum of the two amounts will give the *total move-
ment of each valve on the stem to accomplish the required
range*.   The tangents drawn from point Q to the various
lap circles, and the perpendiculars dropped to them
from the point Q', form precisely the same construction
from Q as a centre that the diagram for the plain valve
did from O as a centre, and these lines with the lap
circles give the relations of the two valves to one an-
other precisely as though the main were a fixed seat
for the cut-off valve.   As in Fig. 15, the distance Oc
represented the velocity of a plain valve at the moment
of cutting off, so in Fig. 68 the distances Qa'', Qb'' repre-
sent the velocity of the cut-off valve relative to the main
valve at the same moment.   In the figure it will be seen
that this velocity is somewhat less than the correspond-
ing velocity for the maximum cut-off by the main valve,

and in fact a somewhat sluggish cut-off is a characteristic
of this valve gear. The only method of quickening this
velocity is to increase the distance between Q and Q' by
increasing the throw or angular advance of the cut-off
eccentric. This, however, increases the diameters of
the lap circles, and the distances which the valves must
be moved on the stem, together with the length of steam-
chest to permit the increased movement. If the full
range of expansion is not required, the entire move
ment on the stem is available for the limited range, and
this can be put to good use by increasing the quickness
of the cutting off ; but generally in practice it is neces-
sary to adjust the conflicting requirements to one an-
other with a view to securing the best compromise
possible.

The greatest distance apart of the centres of the main
and cut-off valves, or in other words the half travel of
the cut-off relative to the main valve, is the distance
QQ'. Should the cut-off plates be brought so near
together that the negative lap equalled this distance
QQ', the cut-off valve would close the main valve port
for an instant and immediately reopen it. Should this
happen before the final cut-off by the main valve, it
would give a readmission of steam. To determine if
this is possible, strike the negative lap circle in ques-
tion with radius QQ'. Draw its tangent c through Q
and a crank line D parallel to the tangent. This crank
line D comes well within the main valve lap circle, indi-
cating that the latest possible cut-off by the cut-off valve
at which the momentary closure occurs takes place after
the closure of the port by the main valve. Had the line
D come tangent to the lap-circle it would have indicat-

ed that the momentary closure of the main valve port
would have occurred just at the closure of the port by
the main valve ; and had it fallen without the lap circle,
as in Fig. 69, it would have indicated that some of the
later grades of expansion would have been accompa-
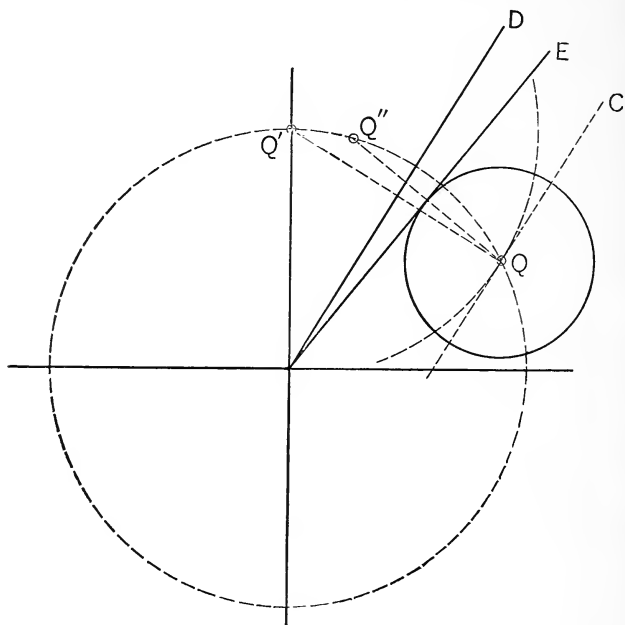
Fig. 69

nied with readmission. This is of but little moment,
as, unlike the previous gear, it affects the late and not
the early cut-offs ; but it can be easily avoided. Thus
in Fig. 69, lines C and D being parallel, QQ', which is
perpendicular to C, is also perpendicular to D. If the

position of Q' be altered to Q'', such that QQ'' is perpendicular to E, it is clear that D and E will coincide, the limiting condition will have been reached, and if
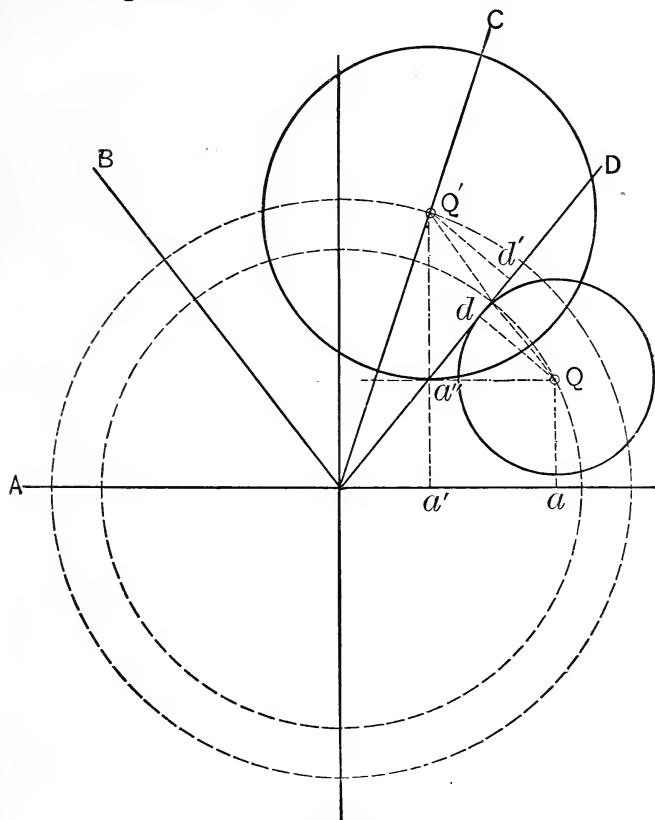
Fig. 70

the centre of the cut-off lap circles be located slightly to the right of Q'', no possible readmission can occur.

One point requiring attention remains. It is clear

that were the cut-off plates quite narrow, they might
when screwed well apart pass entirely over the main
valve ports and readmit steam by their back edges, and
the width must plainly be made great enough to pre-
vent this.   In Fig. 70 place the crank at the dead-point
position A, and determine the lap necessary for cut-off
at that extreme position.   The valves are located at dis-
tances Qa and $Q'a'$ from their mid positions, and $Q'a''$ is
the radius of the required lap circle.   As the crank
mounts upward Qa lengthens and $Q'a'$ shortens, until at
crank position B parallel to QQ', Qa and $Q'a'$ are equal.
Beyond B, Qa exceeds $Q'a'$, until at C, $Q'a'$ vanishes.
Beyond C, $Q'a'$ falls upon the rear side of the crank, and
at D, $Q'a'$ becomes $Q'd'$.   At this point the distance
apart of the centres of the valves is $Q'd'$ plus $Q\,d$.   In
other words, that distance has diminished from $Q'a''$ to
zero, and increased again in the opposite direction to
Qa plus $Q'd'$.   With the crank at A the edge of the
cut-off plate just closed the port, and at D it will have
closed the port by a distance $Q'a''$ plus $Q'd'$ plus Qd;
and the width of the plate must equal this distance,
plus the width of the main valve port, plus an allow-
ance for tightness—say $\frac{1}{4}''$.

THE BUCKEYE VALVE GEAR.

This   exceedingly ingenious valve gear is in one
sense a combination of the two preceding.   To the
small clearance and mechanical capabilities of the Meyer
valve it unites the turning eccentric of the Gonzenbach
with its convenient attachment to the shaft governor.*

* The Buckeye was the pioneer shaft-governor engine.

At the same time it is not limited in range as is the Gonzenbach, and it has a sharper cut-off than either one. Balancing and other features of the valve disguise its relationships somewhat, but discussion of these features is beyond the scope of the present work. The
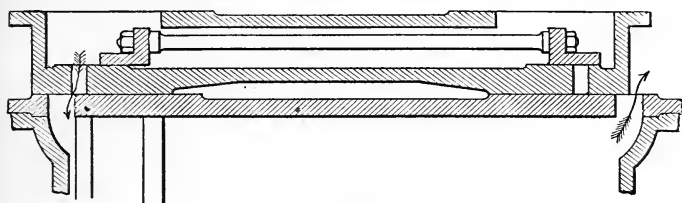
Fig. 71
The Buckeye Valve.

construction of the valve, so far as its action on the ports is concerned, is shown in Fig. 71, from which it will be seen that it takes steam from within its box-like form, and exhausts by its ends into what with other valves is commonly the steam-chest. The cut-off valves are similar to those of the Meyer system, except that they are secured immovably upon the rod.
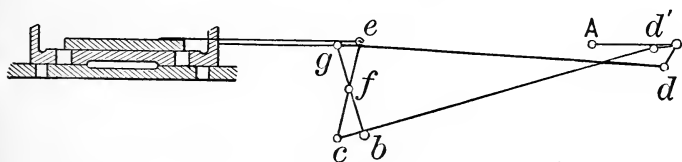

Fig. 72

Fig. 72 is an ideal view of the same valve with a diagram of the eccentrics. The position of the crank being at A, the main eccentric, by reason of the valve exhausting by its outside edges, is at d. The main eccentric and valve rods are connected to a rocker

pivoted at *b*. This rocker does not change the motion
of the main eccentric in transmitting it to the valve.
The cut-off eccentric and valve rods are also connected
to a rocker, the former at *c* and the latter at *e*. This
rocker is pivoted at *f*, which pivot is carried by the
main rocker. It is clear that the motion which the
cut off eccentric rod imparts to the lower end of its
rocker is, if the eccentric be properly set, precisely the
same as that required for a Gonzenbach cut-off valve,
having its seat in line with the eccentric rod. But the
distance *bc* always equals *eg*, or, in other words, the
motion of the upper end of the cut-off rocker relative
to the main rocker is the same as that of the lower end
relative to a fixed valve seat. That is, the motion of
the cut-off valve relative to the main valve is the same
as that of the Gonzenbach cut-off valve relative to its
fixed seat. With the Gonzenbach gear a single set of
ports through the cut-off valve-seat served for both ends
of the cylinder, and it was shown that, in consequence,
there was danger in the early grades of expansion of a
readmission of steam before final closure of the port by
the main valve. With the construction of the present
gear this is avoided, and, properly proportioned, there
can be no readmission in either the early or late grades.
The exhaust taking place at the ends of the main valve
locates the eccentric diametrically opposite from its
usual position, and the cut-off rocker does the same for
the cut-off eccentric. The angular position of the cut-
off eccentric rod also moves the cut-off eccentric from
the positions shown in Fig. 73 by the same angle.
Since the action of the cut-off valve is essentially the
same as in the Gonzenbach gear, it may be represented

by essentially the same diagram as in Fig. 73. Q is as usual the centre of the main valve lap circles and Q', Q'', Q''' of the cut-off (negative) lap circle for different points of cut-off, Q' being the position for cut-off at
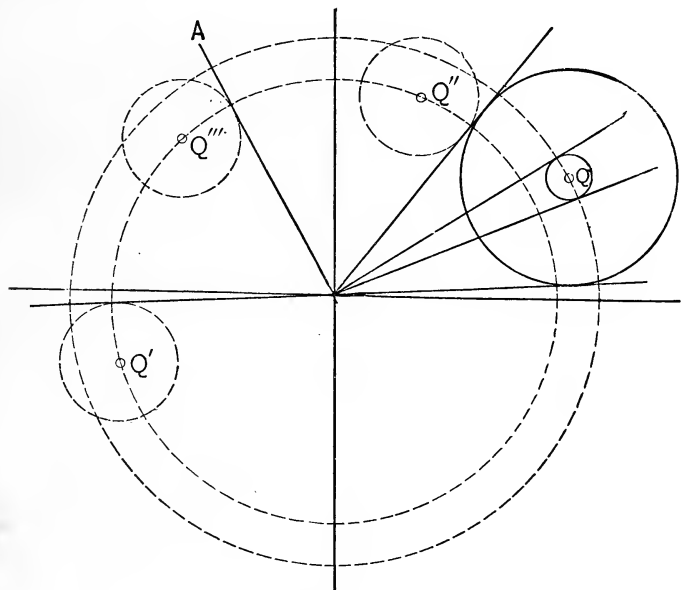

Fig. 73

zero, Q'' for latest cut-off, i.e., at the main valve closure, and Q''' for any desired crank position A. The opening of the ports by the cut-off valves is in this instance of little moment, since it occurs during the previous stroke, when the admission port for the end of the cylinder under consideration is out of action.

THE STRAIGHT LINE INDEPENDENT CUT-OFF GEAR.

Fig. 74 is a horizontal section of a steam cylinder
fitted with the above valves, the bottom of the figure
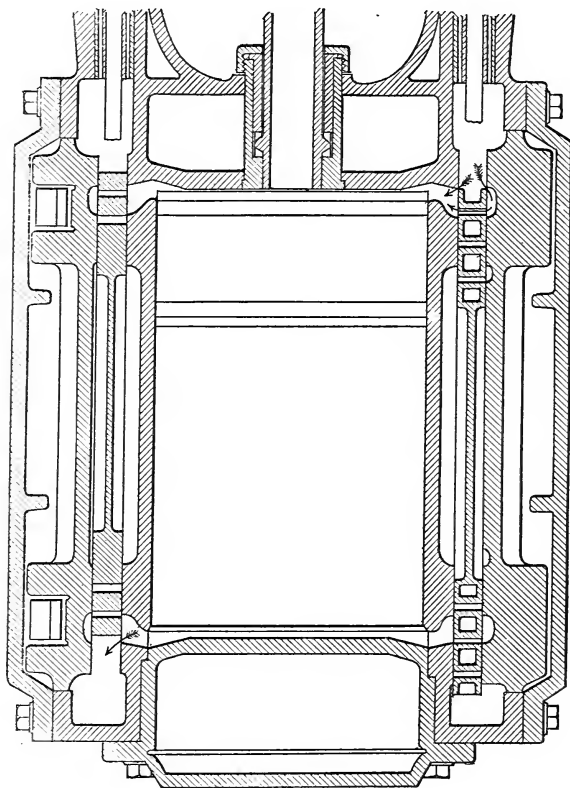
Fig. 74

The Straight Line Independent Cut-off Valve.

showing the steam and the top the exhaust valve.   The
two are of similar construction, both being fitted with

relief plates and multiple ports, and both acting by their outside edges. They differ chiefly in that the exhaust valve is driven by a fixed, and the steam valve by a swinging eccentric. The Bilgram diagram as applied to the gear consists of the diagram already
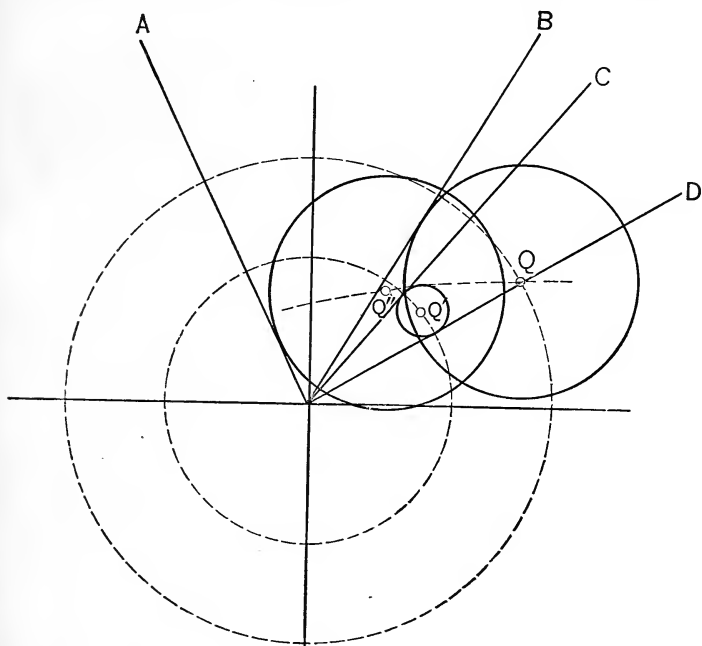
Fig. 75

familiar for the swinging eccentric gear, but with the exhaust lap circle occupying a fixed position instead of the moving one of the usual swinging eccentric gear. Contrary to all previous practice with independent valves, this engine is arranged for positive lead in the

late and negative in the early cut-offs. The object of this is as follows : It is generally understood that compression does not begin to bring the piston to rest until the pressure on the compression side exceeds that on the expansion side of the piston. With an early cut-off this state of affairs occurs at some distance from the end of the stroke, but at later cut-off the expansion curve is higher and the compression curve does not rise so soon to equal it. Hence the effect of the compression in bringing the parts to rest is lessened at late cut-off, and to make up for the deficiency a prominent lead is given.

The application of the Bilgram diagram is shown in Fig. 75, in which Q is the centre of the steam valve lap circle for greatest throw, cut-off at B; Q' is the fixed centre of the exhaust lap circle, and Q'' the centre of the steam circle with the eccentric shifted for cut-off at A, the path of the eccentric centre being QQ''. It will be observed that for cut-off at B the lead is positive and at A negative. The fixed position for compression is C, and for release D.

THE BILGRAM VALVE GEAR.

This gear has been designed to provide a more rapid cutting off than the Gonzenbach or Meyer gear. The following description is from Mr. Bilgram's book on this subject (now out of print):

Both valves are operated by one single eccentric I, Fig. 76; the main valve directly by the eccentric rod, and the cut-off valve through a peculiar mechanism, consisting of four members; viz.: the link, the rocker,
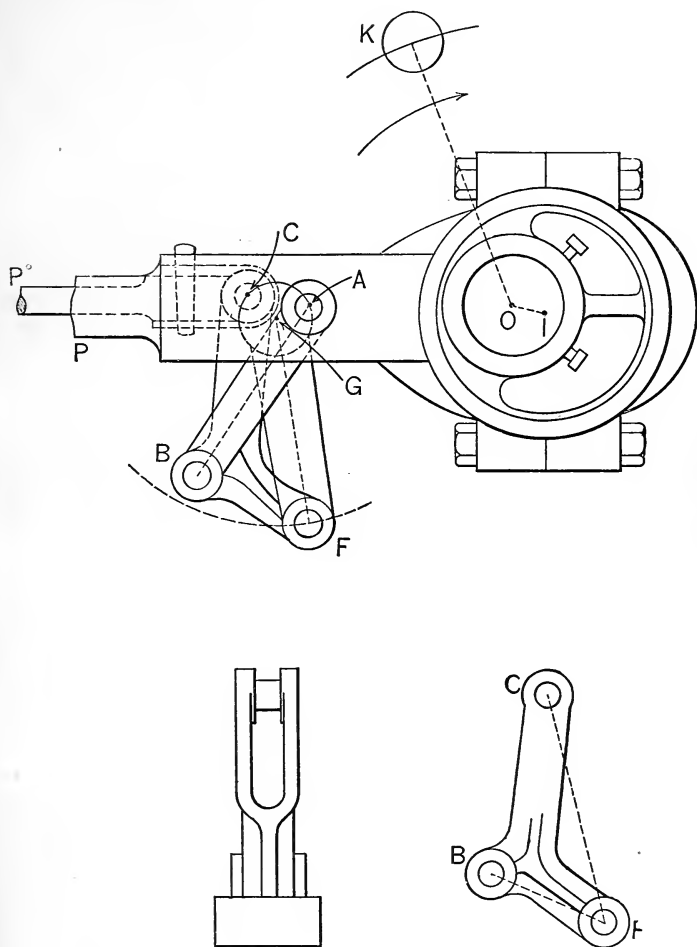
Fig. 76

the cut-off rod, and the adjustment lever. The link
AB being jointed by the pin *A* to the eccentric rod,
imparts to the rocker a rocking motion on its fulcrum
F. The rocker is of a peculiar shape, as shown de-
tached in the cut, but it virtually represents a bell crank
(or angular lever) having an angle *BFC* of about 50°,
the arm *CF* of which is about twice as long as the arm
BF. To the extreme end *C* of this rocker is jointed
the cut-off rod, by which the cut-off valve is moved.
For the purpose of changing the degree of expansion
the fulcrum *F* of the rocker can be moved in a circular
arc, being attached to the end of the adjustment lever
GF, whose fulcrum *G* is a fixed point.

The study of this gear will consist in the investiga-
tion of the movement of the cut-off valve for several
positions of the adjustment lever. In every case we
shall proceed from the neutral position of the rocker
(found by transferring the eccentric rod to the centre
of the crank shaft), remembering that the movement of
the mechanism will be symmetrical to both sides of this
position. Besides, we shall as usual neglect all com-
plicating influences resulting from the angularity of
the several members ; and besides, we shall assume the
movement of the pin *A* to be strictly circular and co-
incident with the movement of the eccentric *I.*

At first we move the adjustment lever until the line
CF of the rocker assumes a vertical position (see Fig.
77), the theory for this position being the least compli-
cated. When the mechanism is in operation, the pin
A will move in a circle, and hence the points *B* and *C*
will move in the arcs *b'b''* and *c'c''*. For the latter arc
we shall substitute the chord to simplify the theory.

When the crank is on its centre, the pin A will occupy
the position $A°$ corresponding to the position of the
eccentric I, and since the link AB represents, as it
were, the eccentric rod for the cut-off gear, we can
measure the angle of advance $δ° = YAA°$ by drawing
AY at right angles to BA. The angle CFB being $50°$
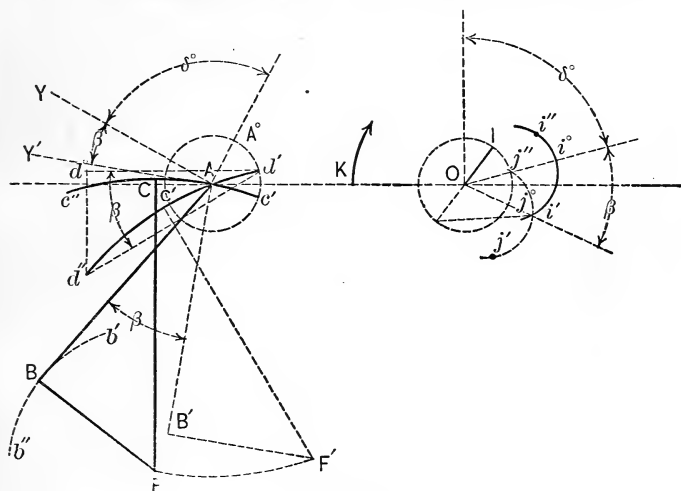
Fig. 77

and AB being at right angles with FB, or approximately
so, it is evident that the angle $YAA° = δ°$ exceeds
the angle of advance of the eccentric I by $50°$. Owing
to the dimensions of the rocker, the travel of the point
C, and consequently also the travel of the cut-off valve,
equals double the travel of the main valve ; and hence
we can find the ideal eccentric $i°$ of the movement of
the cut-off valve for the considered grade by advancing
the line OI through an angle of $50°$ and doubling its
length.

Next we move the fulcrum F towards the right to F' to change the grade, and denote the angular change of the rocker by the Greek letter β. The corresponding angular change of the link AB is practically the same, and the angular advance is consequently farther increased by this angle. We can therefore draw the line Oi', but we have yet to find its length. The movement $d'd''$ of the pin C' of the rocker is doubtless the same as it was before; but being inclined, it is only its horizontal component $d'd^\circ$ that is transmitted to the valve, and the throw Oi' of the ideal eccentric for this grade will be less than Oi°. The necessary reduction can be made by drawing the line $i^\circ i'$ at right angles to Oi', which will be understood when we consider the similarity of the triangles $Oi'i^\circ$ and $d'd^\circ d''$.

In moving the fulcrum F of the rocker in the opposite direction we would have found the ideal eccentric i'', and other positions of the fulcrum F would furnish more points of the locus of ideal eccentrics. The angle $i^\circ i'O$ being a right one, it will easily be understood that all the ideal eccentrics will be located in a circle of which the line Oi° is a diameter.

These results relate to the absolute movement of the valve, and to find the ideal eccentrics for the relative movement we move the locus in the direction of and through a distance equal to IO. Having thus determined the locus $j'j^\circ j''$ of the relative movement, we can draw the valve diagram, Fig. 78, in the usual manner.

This diagram now shows that the cut-off can be adjusted to any point between the crank angles OA and OE as the angular adjustment of the rocker is not limited. It shows, moreover, that this valve gear is distin-

guished by the decided rapidity in cutting off. The closure of the steam passages is very sharp for all grades cutting-off before the half stroke ; for a later cut-off however, this rapidity will get less, until when cutting off at *OE* the rapidity of the cutting off of the cut-off, valve will about equal that of the main valve.
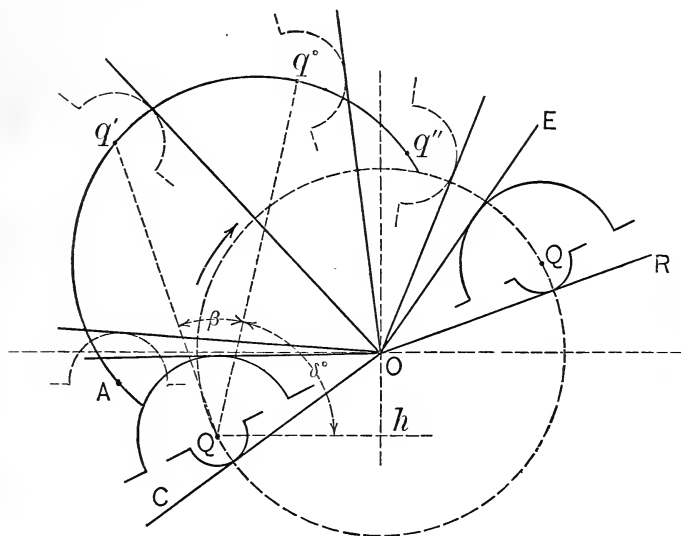

Fig. 78

The rapidity of the late cut-offs can be improved, if desired, by making the arm *CF* of the rocker more than double the length of the arm *BF*, whereby the line *Oi*° will be lengthened, and consequently the locus circle will be enlarged. This change entails an increase of the absolute movement of the cut-off valve above twice the travel of the main valve. Another measure consists

in increasing the negative lap of the cut-off valve; but it should never exceed the positive outside lap of the main valve. A reduction of the angle *BFC* of the rocker would likewise be efficient, but this reduction is attended by an increase of certain irregularities.

The sharpness of the cut-off will in reality be slightly less than indicated in the diagram, from the fact that the movement of the pin *A* is not circular, as was assumed, but is more or less flattened.

The proportioning of the mechanism requires some
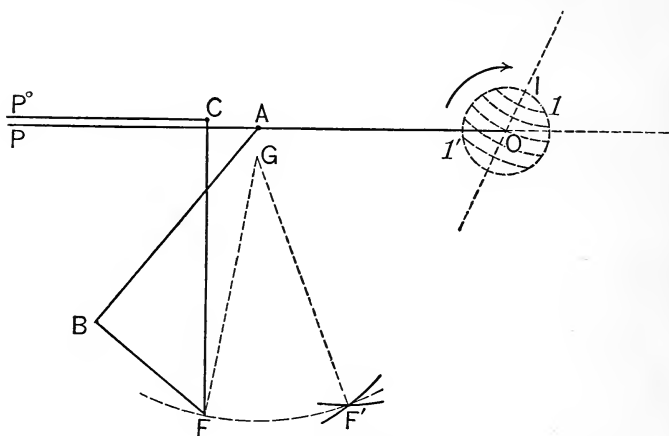
Fig. 79

care, for on it depends largely the proper operations and regularity of the cut-off. To this end we draw the rocker *BFC* (Fig. 79) with the line *CF* in a vertical position and the line *BF* at an angle of 50°, and make the arm *BF* from three to four times the throw of the eccentric, and the arm *CF* twice as long. (The figures

given have been tested by a number of experiments.)
Then we draw the link *AB* at right angles to *BF* and
make it about $\frac{6}{7}$ of the length of *CF*. The eccentric
rod *OP* can next be shown in its neutral position, pass-
ing through the end *A* of the link. The cut-off rod
CP° can likewise be shown.

To find the length and position of the adjustment
lever *GF*, it is necessary to make a model of thin wood
or veneer, consisting of the eccentric rod, the link, the
rocker, and the cut-off rod. Next we draw the orbit of
the eccentric, and on it the exact position of the eccen-
tric, say for every one sixth of the stroke of the piston,
which may be done in the following way, identifying
the eccentric path with the path of the crank pin :

We divide the diameter of the orbit passing through
the initial position of the eccentric *I* in six equal parts,
and draw the projection arcs of the proper radius
through those points as shown. The next thing to be
done is to attach the model to the drawing by joining
the parts properly together with pins or thumb tacks,
and fastening the ends *P* and *P°* of the rods to two
slides representing the valves. The right end of the
eccentric rod may be provided with a needle point
which at first we put in the centre *O*, when we set the
rocker directly over the position shown in the drawing,
and mark the relative position of the two slides repre-
senting the valves. Then we make two additional
marks on one of them, at a distance equal to the as-
sumed negative lap of the cut-off valve, to show the
relative position for the cutting off on either side.
Suppose now we desire to find the proper position of
the rocker for cutting off at the point 1. To this end

we set the needle point of the eccentric rod in the point 1 of the fore stroke, and fix the valves for cutting off at the proper side, when we will find that the end F of the rocker cannot be moved but in a certain curve. This curve we mark on the drawing by setting a needle point into the rocker and making a slight scratch on the paper. Thereupon we attach the eccentric to the point $1'$ of the return stroke, fix the cut-off valve to the point of closure of the other passage of the main valve, and mark another curve by the point F of the rocker. The juncture F' of the curves must of necessity be the exact position of the fulcrum of the rocker when we desire to cut off at one sixth of the stroke. In this way we can find the required position of the fulcrum for all the other grades, which will be found to form a curve. By substituting a circular arc for this curve, osculating as closely as possible, we obtain the location and length of the adjustment lever GF. An arc can generally be found to agree with the constructed curve with an almost absolute precision ; and hence it will be seen that this valve gear will admit of a practically perfect equalization of the difference between fore and return stroke.

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